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PHOTOPROJECTION—NEW CHALLENGE TO SHIPBUILDING TRADITION.

BY ALBERT F. BIRD.

Second Prize Winner in the 1947 Prize Essay Contest.

Shipmodels, centuries old, preserved in bottles fascinate mariners and shipbuilders. While there is always the speculation as to how they were committed to the glass, one point is undisputed; the ship cannot be gotten out again.

There is another way in which a ship can be preserved in glass which has the advantage of easy removal—not only in its original small scale, but full size. Of widespread potential usefulness in wartime, the process disclosed in the following pages permits design, layout and fabricating patterns to be made during peace time and stored. When need for the ship is urgent, patterns ready for fabrication of steel components can be available in a few hours' time. The traditional sequence of shipbuilding is shortened by the elimination of the mold loft and the many full

size layout operations connected with it. Photoprojection and photolofting make this possible.

Traditional Mold Lofting Means Slow Start.

Mold lofting is one of the sacred shipbuilding traditions because from the loft come hundreds of wooden templates to be used in cutting and forming of hull steel. To the mold loft comes the designer's table of offsets—many columns of figures from which the hull body plan is laid down. Crouching on a wide expanse of smooth floor, loftsmen "lay down" the ship. Full size, longitudinal plans of waterlines, buttocks and diagonals are faired with thwartship body plans until complete fairness in all planes is established.

Once a fair body plan is reached, templates of plating, frames, longitudinals, floors and the other structural members are laid out and constructed of wood.

This process for the Landing Craft Infantry, Large, LCI(L) during World War II consumed about 5000 man hours. These hours were spent when the need for a speedy start in shipbuilding was critical. Each hour in this stage of the game was doubly important. No steel could cut until templates were made.

Aside from the delay when no time is available mold lofting involves certain inherent drawbacks. First, when many ships are built to one design by different yards, each yard traditionally prepares its own mold loft, does its own lofting and template layout. If twelve yards build to a single design 60,000 man hours are spent in lofting and template layout in an LCI(L) program. Thus time which might be saved is lost in each yard involved.

Second, mold lofting as practiced does not promote standardization in structure. Sub-assemblies made from templates lofted by Yard A cannot be expected to fit with similar parts to templates laid out by Yard B. Each yard can produce a ship, but components are not standardized between yards. These discrepancies are unavoidable under present lofting methods, but can be eliminated by the procedure to be described.

Third, the mold loft itself is an expensive feature covering thousands of square feet of floor space, requiring a major capital outlay to create. Its elimination would add to each shipyard valuable working space or would represent a significant saving in building cost.

Aircraft Industry Sets New Lofting Pattern.

Originally the aircraft industry adopted shipyard mold lofting, actually hiring shipyard people to carry on the work. Impatient to improve on traditional layout methods, the aircraft industry outgrew the laborious and approximate mold loft techniques. In 1938 the Glenn L. Martin Aircraft Company inaugurated a technique for laying out aircraft parts known as "Photolofting." This method rapidly replaced the mold loft until today the mold loft is regarded as merely an interesting piece of historical backgound. Photolofting exceeds mold lofting from the standpoint of accuracy, speed and cost.

Large size craft are produced by photolofting methods. B-29 bombers were built on this basis by widely scattered sub-assembly fabricators. Infinite standardization was achieved so that a very few assembly plants joined the components from a multitude of sources into the completed airplane. All metal was cut from photolofted templates. Photolofting is not experimental in the aircraft industry. It is an accepted and proven tool of standard manufacture.

Production of Phototemplates.

Drawing for photolofting and photoprojection is specialized to a considerable degree. Precision beyond that normally required in shipdrawing is achieved by several highly developed procedures. The more important considerations include a stable drawing medium, mechanically ruled grid lines, attention to width of line, fine tolerance in all dimensions and drawing to a large scale usually 3″ equals 1′.

Metal sheet, usually aluminum, suitably lacquered and surfaced, is a good drawing medium. The pale green non-reflecting surface commonly given the sheet makes a sharp contrast with pencil lines. Eyestrain is thus held to a minimum and the plate is a favorable photographic object. Unlike paper or cloth the metallic drawing is stable in varying degrees of temperature and humidity. Drawings do not change from year to year, hence precision in the photographic process is assured. Photographic limitations and ease of handling have resulted in a maximum sheet size of five feet by nine feet, although sheets under this size are frequent, since photographic apparatus is widely adjustable.

Mechanical ruling of reference lines on the metal sheet establishes precision datum points. Such lines assist in registering adjoining sheets of a single drawing. Human errors in ruling long lines are held to a minimum.

Drawing is done with hard pencils kept to a carefully controlled sharpness. A sharp dense line is made having a thickness not exceeding .005″. Too light a line will lose sharpness in reproduction, while a line too wide becomes intolerable when photo enlargement takes place.

Scales and instruments are calibrated in fine intervals to permit extreme dimensional accuracy. Tolerances are held to .005″ per foot so that even with a fourfold blowup precision is within permissible limits.

The scale used in shipbuilding photolofting, 3″ = 1′, is well suited for several reasons. First, vessels under two hundred feet in length can be drawn to this scale on ordinary drafting room equipment. Standard metal lofts placed edge to edge on tables can take waterline and body plans without using excessive floor space. Contracted station intervals in waterline plans reduce space requirements still more. Second, full size blow up (four times drawing size) by the camera gives a line width very comparable with the ordinary mold loft pencil line. Third, this scale permits adequate precision in drafting. Smaller scale than 3″ = 1′ introduces unavoidable errors in enlargement, larger scale is unnecessary for accuracy and causes wasteful use of space.

Drawing for photolofting is done by a draftsman schooled in mold loft practise. This combination of skills in a single man is not widely found at the present time. Either a man is an excellent ship draftsman without layout and lofting knowledge, or he is a mold loftsman skilled in the tricks and traditions of his trade but not schooled in the science and techniques of the drafting room. In order to realize the greatest benefit from photolofting it is necessary to develop the combination of drawing and lofting skills in the same individual. This training is not a simple operation because the complete dependence of the shipbuilders on this man requires completely reliable results. As will develop later in this paper, adequate training of drafting loftsmen is an indispensible part of the preparedness represented by photolofting.

Let us follow the path of the LCI(L) design by photolofting from the conception of the vessel to a finished template, see plate

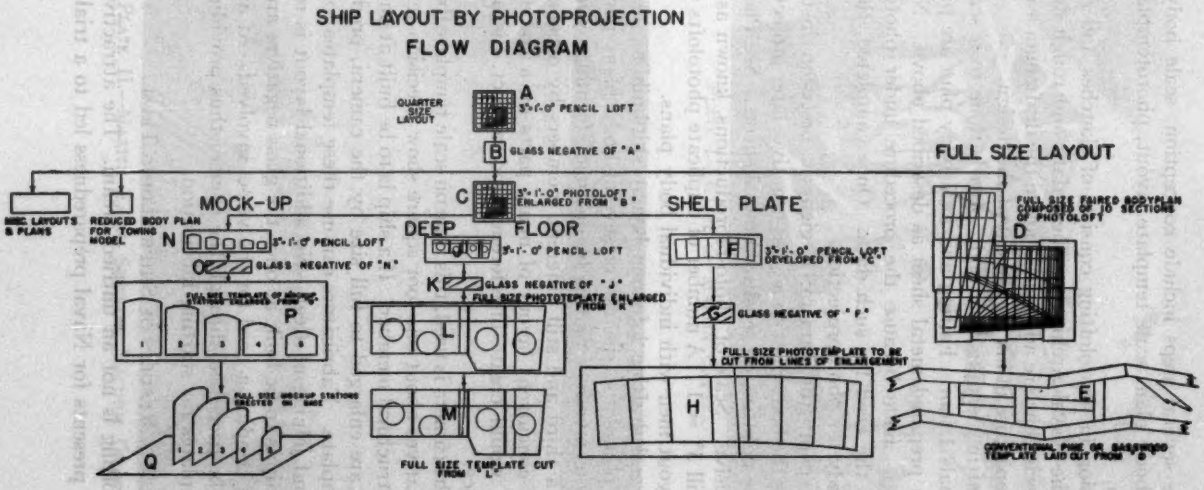


PLATE XIV.—PHOTOPROJECTION FLOW CHART FOR SHIPBUILDING.

XIV. The several steps include conception, scale body plan, photoloft body plan, scale template layout, phototemplate and fabrication.

Conception in photolofting consists of sketches and descriptions by the Naval Architect. These data and rough drawings are not precise but are adequate to give the draftsman a reliable point of departure in his refinement of the design.

Ship drafting leading to a body plan is done to a scale of 3 inches equals 1 foot. Fairing and final body plan are precision drawn, on prepared metal sheet as described above. No table of offsets is made because the procedure under photolofting eliminates the need for such data. Once completed, the body plan is ready for photoprojection.

At this point the camera, a gigantic precision instrument reproduces the body plan on a glass negative, later projecting the lines of the negative on sensitized metal sheets. See Plates I to VIII inclusive. Scale of photo-reproductions, known as photolofts, is still 3″ = 1′. A number of duplicate photolofts provide several layout men with individual body plans.

The several drafting loftsmen layout shipbuilding templates from the photolofts, the use of several body plans permitting simultaneous layout of plating by one man, framing by another, floors by a third and still different members by others. Note that such concurrent layout of many parts on a conventional mold loft would create interference and conflict, discouraging the practice.

Again the camera is used, this time on scale template drawings. Glass negatives about one foot square show precision templates of every structural member. If a ship is to be built at once these negatives are enlarged to full size by the camera, producing a photo-template. Fabricators can use these templates for actual cutting and joining of steel. No additional layout is required.

If no ship is to be built at once the glass negatives are stored in a vault pending a need for the vessel involved. At any time they can be processed through the camera thus providing complete templates for starting fabrication.

Results of Shipbuilding Trial.

Photolofting is not an untried dream. The attractive possibilities it presents for Naval preparedness led to a trial project

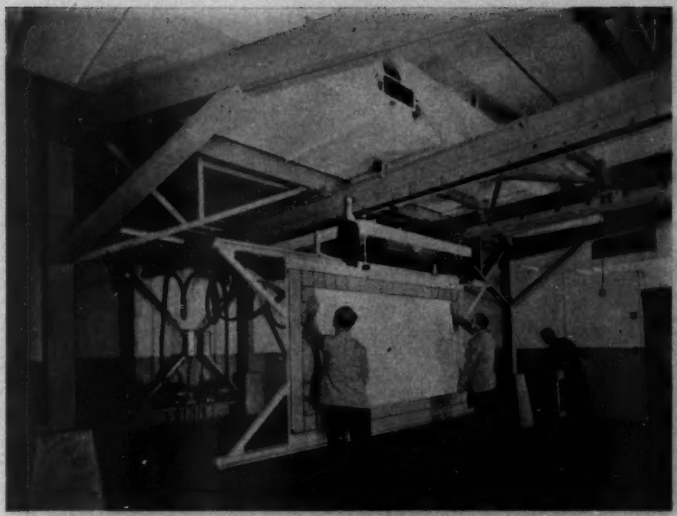


PLATE I.—EASEL OF MARTIN CAMERA.

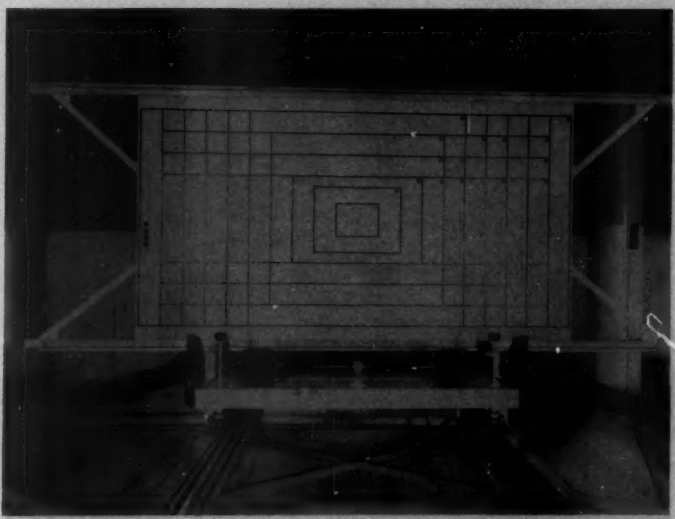


PLATE II.—MARTIN EASEL SHOWING VACUUM GROOVES.

PLATE III.—THE ROOM BEHIND THE LENS.

PLATE IV.—PHOTOGRAPHER READY TO DARKEN ROOM.

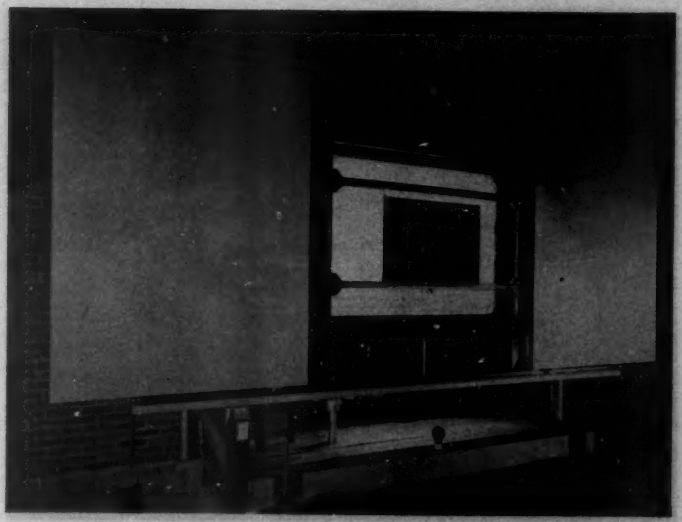


PLATE V.—LIGHTS BEHIND NEGATIVE PROJECT LINES.

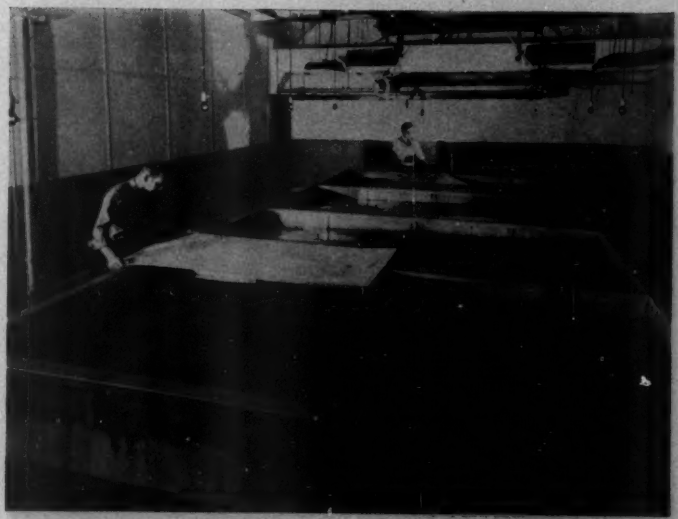


PLATE VI.—PHOTOLOFTS ARE DEVELOPED AND DRIED.

PLATE VII.—CAREFUL CHECKING ASSURES ACCURACY.

PLATE VIII.—PHOTOLOFT FOR TOOL AND JIG MAKING.

PLATE IX.—LC1 (L) FOREBODY PLAN 3″ = 1′ ORIGINAL AND PHOTOLOFT.

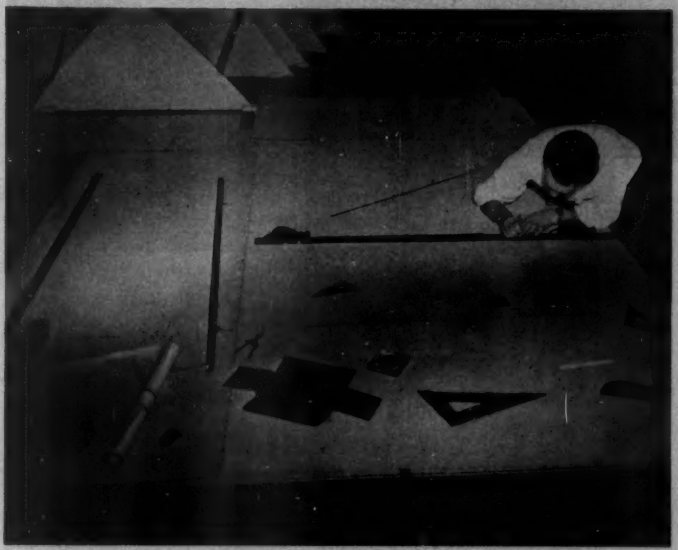


PLATE X.—PRECISION DRAFTING OF LC1 (L).

PLATE XI.—FULL SIZE PHOTOLOFTED FOREBODY PLAN LC1 (L) ASSEMBLED IN SHIPYARD.

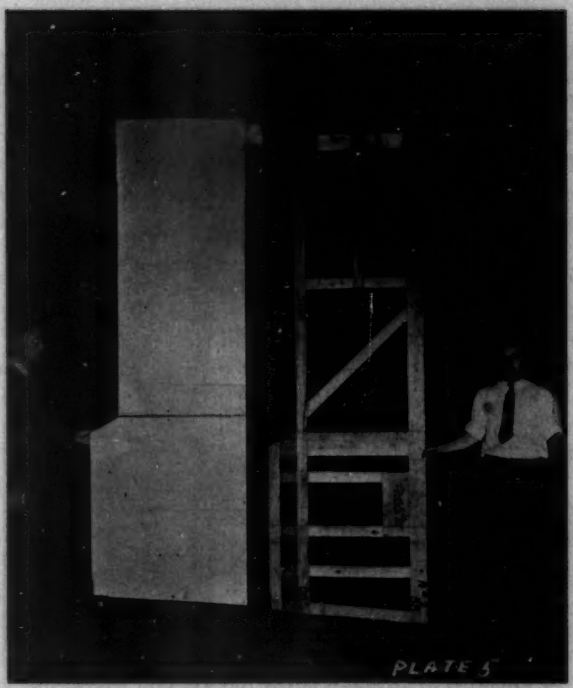


PLATE XII.—PHOTOTEMPLATE COMPARED WITH ORIGINAL PINE TEMPLATE.

PLATE XIII.—PINE TEMPLATE SUPERIMPOSED ON PHOTO TEMPLATE.

using the LCI(L) as an example. The procedure outlined above was followed by an experienced shipbuilder and photo-templates resulted which were even superior in clarity and precision to the comparable pine templates used for actual LCI(L) construction. See Plates IX to XIII inclusive.

Plates XII and XIII show the comparison between templates made by conventional and photoloft methods. Discrepancies were rare, in fact where differences were found, the mold loft proved to have changed its shape since offsets and templates were made. Photolofting was declared to be a practical and accurate method of template production.

Naval Use of Photolofting.

The following criteria determine whether or not photolofting should be used to "preserve" ships.

a. Photolofting finds its best field where *numerous* one design vessels are to be built. Cost and layout time on a single ship do not show attractive advantages for photolofting.

b. Leading and following yard organizations, common in World War II, offer the best possible field for use of photo-templates. Under actual mold loft layout conditions lines were laid down and faired by numerous following yards. Templates for identical structural members were produced separately by each of dozens of builders. Time was thus lost which might have been saved.

c. Desire for a speedy start in ship construction is the most potent reason for adoption of photolofting. By this means the Navy can design and lay out ships when there is plenty of time, but can nevertheless, rush them into production with no lost motion in an emergency. Design and photo-template making are carried up to the ship building stage save for the last photographic process.

Conclusion.

Here is a weapon; a way to preserve a ship in glass. Quicker shipbuilding can result. Photo-templates of battle damage hull parts could be quickly made and flown to forward repair areas. Skill in layout and lofting would no longer be necessary in shipyards. Infinite standardization of hulls would be unavoidable.

This weapon cannot be created instantly when the emergency is upon us, but should be developed carefully while there is time. Drafting loftsmen should be trained, camera techniques should be perfected, photolofted ships should be built. The weapon should be kept thus sharpened for the day of need.

REFERENCES.

1. Bureau of Aeronautics Drafting Manual, "Lofting and Layout—Use of the Loft." Manual No. 19.

2. Bureau of Aeronautics Manual, "Templates and Layout—Reproduction of Lines, Part 1 and Part 2"; Manuals No. 14 and 15.

3. BUSHIPS Contract NObs-22609.

Acknowledgment is made to Glenn L. Martin Aircraft Co. and to George Lawley and Sons Corp. for assistance and inspiration in the development of Photolofting for Shipbuilding.

TURBO CHARGING AND GAS TURBINES.

By Dr. Alfred J. Buchi, Winterthur, Switzerland.[1]

1. Introduction.

When I was asked by your very highly esteemed Head Engineer and Diesel engine expert, Dr. Magdeburger, whether I could give you a lecture, I hesitated first a little to agree to do so. Being, for the first time after World War II, on a tour to visit my Diesel engine colleagues in your country, I did not feel too well prepared to give you a very accurate picture of the present position and possibilities of all the Diesel and other prime mover engines built and developed at present. This could also not be done in a relatively short lecture. I must therefore apologize that I am only able to speak to you about some outstanding achievements and some newer developments with which I am more personally connected. But our research work includes or touches at least more or less all the newly-developed prime movers. I hope therefore that you will still get some information about the prospects of how the more and more needed mechanical power can still be generated better and more economically. Let me begin with

2. Some Historical Reminiscences About Turbocharged Diesel Engines.

It might be worth while to remember that as far back as 1911 the first turbocharged Diesel engine had been built in Switzerland at the Sulzer Works. It was thoroughly tested up to a maximum charging pressure ratio which till now—I am sorry to say—has not been generally applied to this kind of engine. These tests have shown that turbocharging promises excellent results. A BMEP up to 228.5 psi. with a charging pressure of 28.4 psig. was generated. With the then used air blast fuel injection, the increase in output compared with the normal Diesel engine of the same cylinder size was 3.575 times more.

1. Lecture given at the U. S. Navy Department, Washington, D. C., on June 9, 1947 (Auditorium of Dept. of Interior).

Figure 1 shows the main test results in function of the varying charging pressure ratios. On this engine as well as on very many others built later, cooling of the charging air after the charger was used. Because of the decent increase in temperature during pre-compression, that was of course very effective. The engine output was increased from 20 Bhp. (normal Diesel engine with atmospheric charging pressure) up to 71.5 Bhp. Further on the fuel

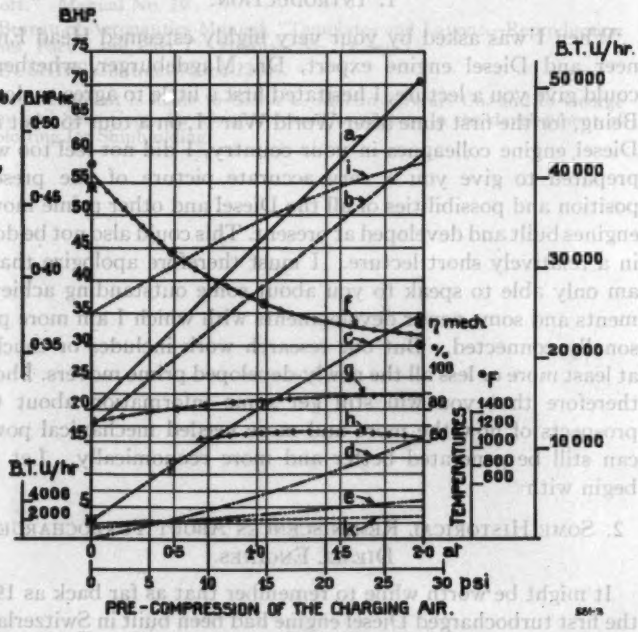


FIGURE 1.—TEST RESULTS OF THE FIRST HIGH-PRESSURE SINGLE-CYLINDER FOUR-CYCLE BUCHI-TURBOCHARGED DIESEL ENGINE OF 1911. CYLINDER BORE 8⅔", STROKE 13¾", 300 RPM.

a—Total effective work of turbocharged Diesel engine.
b— } Positive work { Diesel engine
c— } { gas turbine
d— } Negative work { turbocharger
e— } { injection air pump
f—Fuel consumption.
g—Mechanical efficiency of Diesel engine.
h—Exhaust temperature behind exhaust valve.
i—Heat losses (piston, cylinders and cylinder cover of Diesel engine).
k—Heat losses of cooled exhaust valve.
N—Fuel consumption of an ordinary, unturbocharged Diesel engine with the same swept cylinder volume and a normal output of 20 BHP.

consumption dropped quite a lot, from .471 to .364 lb/Bhp./hr., i.e., 23%. The mechanical efficiency of the Diesel engine was increased from 72% to 88%. The heat transfer through the cylinder walls of the engine into the cooling water did not increase by far in proportion to the increase in output. The heat losses per Bhp/hr. diminished therefore quite substantially and improved the economy. It could be confirmed that the many disapproving critics—opposing the new ideas—failed completely with most of their rather inexcusable prophecies.

Before World War I the willingness to take up new technical inventions and, I am also sorry to say, the capability to understand and judge them thoroughly, was not always existant with the captains of the industry. The present, more progressive time in which farsighted men have developed and are developing the atomic bomb, radar, the turbo-jet and turbo-prop-jet planes, the gas turbine, etc., differs quite distinctly from this more conservative but also quieter period.

3. Some Outstanding Achievements in Turbocharged Diesel Engines.

In this short chapter I will refer mostly to European-built engines. This is probably more interesting to you. I assume that most of you know what has been achieved in your country in which today—I am glad to say—most of the turbocharged Diesel engines are now built.

(*a*) Figure 2 shows test results of a 6-cyl. MAN engine of 11¾″ bore, 15″ stroke and 700 Rpm. You see that a Bmep. of 161 psi is generated. The mean piston speed is 1743 ft/min. The charging pressure is, with less than 7 psi at this output, very small because the engine works with ample scavenging and divided exhaust manifolds. The turbocharger turns therefore also with a moderate and reliable speed.

(*b*) Turbocharging of small automotive engines is also well possible. Corresponding test results are given in Figure 3 of a 6-cylinder engine of 4⅝″ bore, 6″ stroke operating between 800 and 1800 Rpm. With the low charging pressure of a little over 3 psi, a maximum Bmep. of 147 psi could be obtained, when the engine was running at the reduced speed of 1175 Rpm.

(*c*) Increase in output for a given cylinder dimension means, of course, reduction in weight. Consequently, all turbocharged

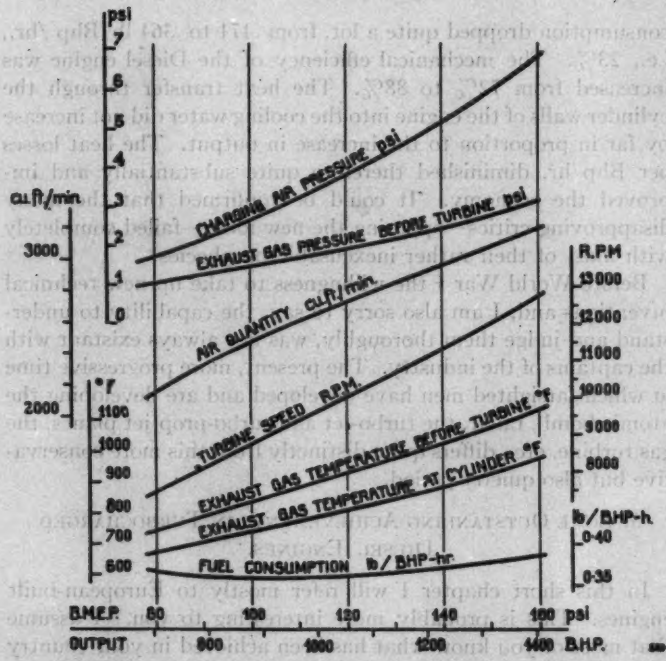


FIGURE 2.—TEST RESULTS OF A BUCHI-TURBOCHARGED 6-CYLINDER MAN-FOUR-CYCLE DIESEL ENGINE, CYLINDER BORE 11", STROKE 15", 700 RPM.

Diesel engines of a given output are much lighter and of smaller dimensions than ordinary engines working with the same cycle.

How far an engine can be reduced in both these respects is clearly shown in Figure 4. This photo shows a 700 Bhp Maybach V-type turbocharged engine. The speed of the engine is 1500 Rpm. and its weight only 8 lb/Bhp. Werkspoor, Amsterdam, has built and is at present building quite a large number of these engines. They are used for driving railcar trains. The Dutch Railway Company runs these relatively short trains in short intervals, and has therefore greatly improved the competitiveness of its railway service.

(*d*) Figure 5 shows a photograph of a three-car Diesel train. It is worthy of note that this German light-weight train holds the world record in railway train speed of over 134.5 miles/hr. Two Maybach engines, one on each end, are installed.

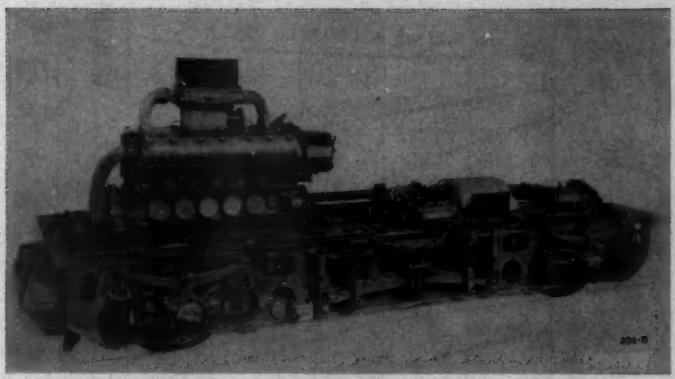


FIGURE 4.—700 BHP BUCHI-TURBOCHARGED 12-CYLINDER TWIN-BANK MAYBACH-DIESEL ENGINE, CYLINDER BORE 6.3", STROKE 7.875", 1500 RPM

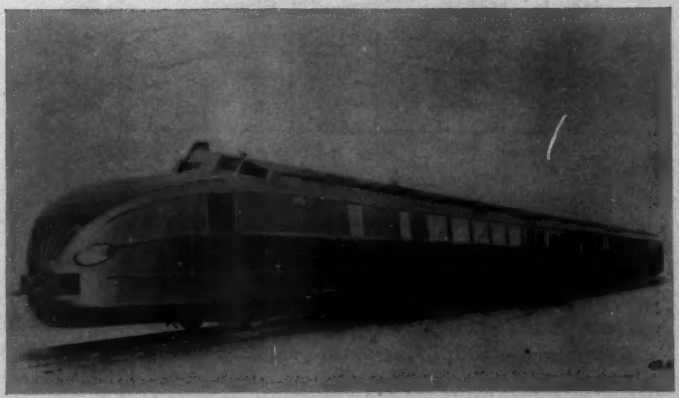


FIGURE 5.—LIGHT-WEIGHT THREE-CAR TRAIN WITH TWO 700 BHP BUCHI-TURBOCHARGED 12-CYLINDER MAYBACH-DIESEL ENGINES. WORLD'S SPEED RECORD FOR RAIL TRACTION OF 134.5 MILES/HR.

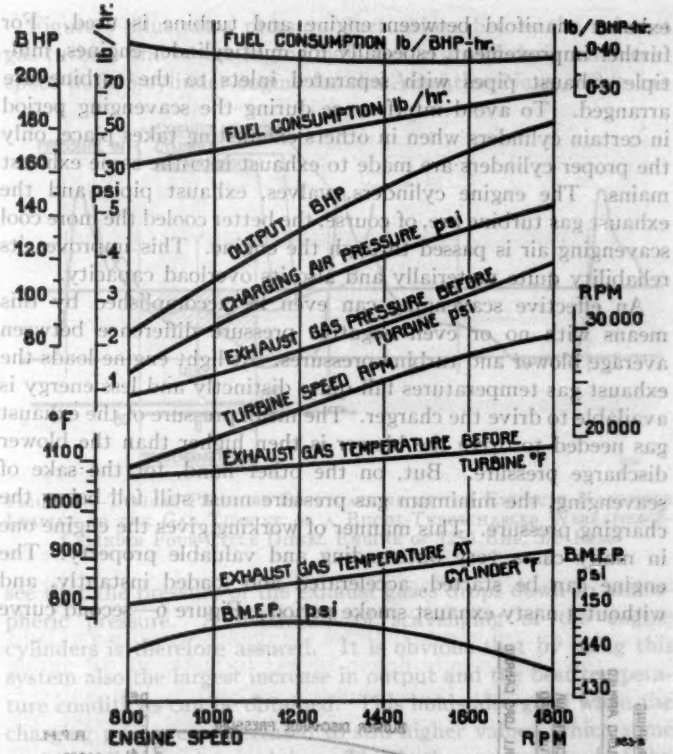


FIGURE 3.—TEST RESULTS OF A BUCHI-TURBOCHARGED 6-CYLINDER FOUR-CYCLE AUTOMOTIVE HIGH-SPEED DIESEL ENGINE, CYLINDER BORE, 4⅝", STROKE 6", 1800 RPM.

The outstanding results of the, so to say, now generally adopted turbocharging system of four-cycle Diesel engines are, first of all, based on the working with very pronounced pressure fluctuations (waves) in the exhaust gas conduits between the engine and the exhaust gas turbine. The pressure peaks and the pressure lows, the amplitudes of these fluctuations, are specially built up and timed. The maximum amplitude is higher than the blower, i.e., the charging pressure. The pressure minimum lies, in contrast, as much as possible below the charging pressure. This special manner of working includes also the case when only one single

exhaust manifold between engine and turbine is used. For further improvement, especially for multicylinder engines, multiple exhaust pipes with separated inlets to the turbine are arranged. To avoid interference during the scavenging period in certain cylinders when in others exhausting takes place, only the proper cylinders are made to exhaust into the same exhaust mains. The engine cylinders, valves, exhaust pipes and the exhaust gas turbine are, of course, the better cooled the more cool scavenging air is passed through the engine. This improves its reliability quite materially and also its overload capacity.

An effective scavenging can even be accomplished by this means with no or even negative pressure difference between average blower and turbine pressures. At light engine loads the exhaust gas temperatures fall quite distinctly and less energy is available to drive the charger. The mean pressure of the exhaust gas needed to drive the blower is then higher than the blower discharge pressure. But, on the other hand, for the sake of scavenging, the minimum gas pressure must still fall below the charging pressure. This manner of working gives the engine one in many cases very outstanding and valuable property. The engine can be started, accelerated and loaded instantly, and without a nasty exhaust smoke period. Figure 6—second curve

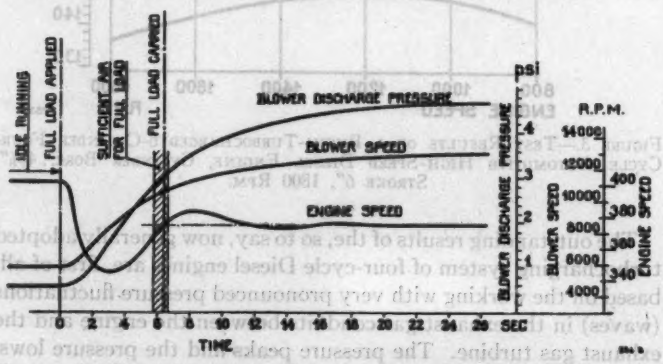


FIGURE 6.—CHARACTERISTICAL DIAGRAM OF A SUDDEN 100 PER CENT LOADING OF A BUCHI-TURBOCHARGED DIESEL ENGINE.

from top—shows the rapid acceleration of the turbocharger and the other items when 100% of the load is suddenly thrown on the engine.

Figure 7 illustrates pressure fluctuations before the exhaust gas turbine and in the engine cylinders in a well-designed Werkspoor-built 8-cylinder engine for the Argentine Navy. One can

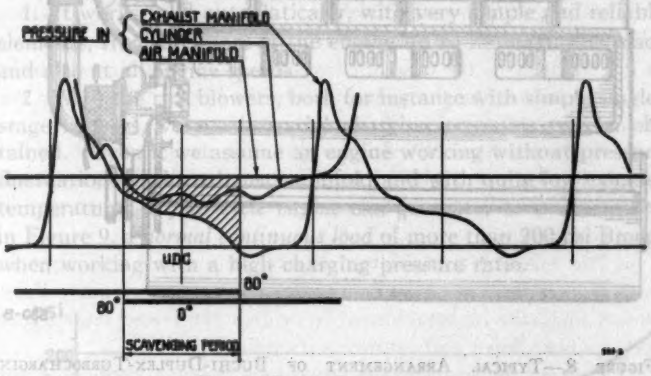


FIGURE 7.—TYPICAL PRESSURE FLUCTUATIONS IN THE EXHAUST MANIFOLDS LEADING TO THE GAS TURBINE OF A BUCHI-TURBOCHARGED WERKSPOOR-8-CYLINDER FOUR-CYCLE DIESEL ENGINE OF 1250 BHP, 375 RPM.

see that the pressure of the exhaust gases drops down to atmospheric pressure. A maximum of scavenging of the engine cylinders is therefore assured. It is obvious that by using this system also the largest increase in output and the best temperature conditions can be obtained. This holds also good when the charging pressure is increased to still higher values, which some engine builders are now doing. This is the general trend of the present development in Europe and here.

4. How Can Still Much More Substantial Increase In Output with Turbocharging Be Obtained?

Such a further development must be safe and should also add some quite desirable new advantages. Therefore, we have, in our research work, investigated according to this principal idea. The most effective improvement we could find was a solution where two blowers are used. Figure 8 shows it in principle. The first blower, advantageously of the centrifugal or axial type, is driven by the Diesel engine itself or, for instance in a ship, from some other power source such as an electric motor. It acts, built as simple as possible more like a fan, as the first

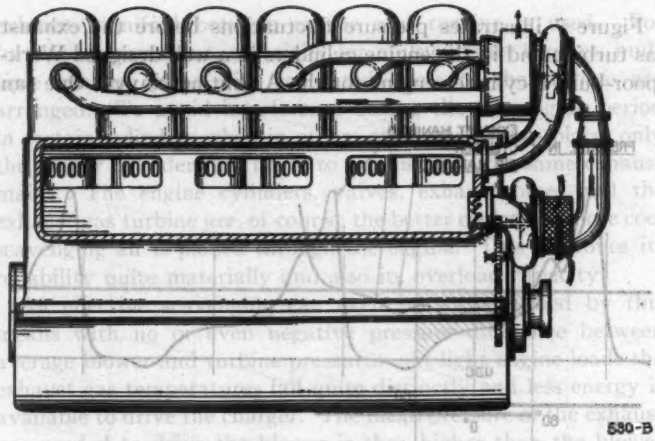


FIGURE 8.—TYPICAL ARRANGEMENT OF BUCHI-DUPLEX-TURBOCHARGING SYSTEM FOR TWO-CYCLE DIESEL ENGINES.

stage of the air charging system. As second stage, follows an exhaust gas-driven turbocharger which is therefore of relatively reduced dimensions and weight. To ensure good starting of the Diesel engine and also to improve the running condition at light loads, the mechanically driven blower furnishes its air during starting and light loads—through non-return flap valves—*directly* to the Diesel engine. On higher loads, however, the mechanically driven blower delivers its pressure air to the exhaust turbocharger. This latter feeds then all the charging air to the Diesel engine.

An irreproachable starting and running of the Diesel engine under all load conditions and speeds, without the necessity of any interference of the engine operator, is therefore still better achieved with this *Duplex Turbocharging System.* Simple non-return flap valves which operate fully automatically are used. The low pressure blower exit is also constantly connected to the blower inlet of the turbocharger. Once that latter blower has, after the start and loading of the engine, been accelerated to a speed where it produces a higher air pressure than the mechanically driven blower, then the above-mentioned simple flap valves will close quite automatically. All the air is then compressed in

series by both blowers acting as air-compounded units. The choice of the correct air manoeuvering is taken care of without any interference by the engine operator. This system offers the following advantages:

1. It works fully automatically, with very simple and reliable elements, from the start of the engine up to its maximum load, and also at all engine speeds.

2. With the two blowers, both for instance with simple single-stage runners, very substantial charging pressures can be obtained. Even if we assume an engine working without pressure fluctuations in the exhaust manifold and with quite low exhaust temperatures, a *four-cycle engine* can generate, as is illustrated in Figure 9, a *normal continuous load* of more than 200 psi Bmep. when working with a high charging pressure ratio.

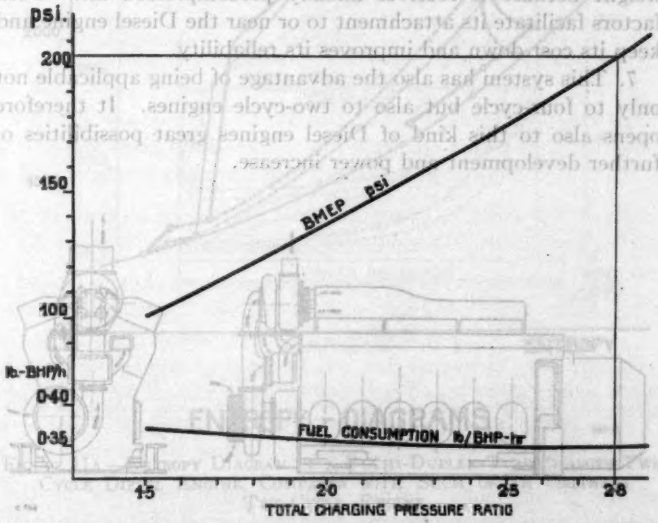


FIGURE 9.—BMEP OBTAINABLE WITH BUCHI-DUPLEX-TURBOCHARGING OF FOUR-CYCLE DIESEL ENGINES, WHEN WORKING WITHOUT SCAVENGING, BUT WITH INTERCOOLER.

Cooling of the charging air is assumed, but no scavenging of the engine cylinders. Highly increased scavenging through the use of exhaust gas pressure fluctuations and divided exhaust

manifolds can also here be applied. This increases the output and improves the reliability quite a good deal further.

3. The "duplex" system has the further advantage that during starting and at light loads there is always a substantial charging and scavenging pressure existing. This is especially a great advantage for engines which must run during extended periods at low loads, as is the case, for instance, in certain Navy vessels.

4. The higher the charging pressure the higher will be also the admissible back pressure behind the turbine.

5. If the mechanically driven blower is designed with radial runner blades and a blade-less diffusor, the Diesel engine itself can be reversed without needing any reversing device on any of the two blowers.

6. The exhaust turboblower is relatively small and light in weight because it receives already precompressed air. Both factors facilitate its attachment to or near the Diesel engine and keep its cost down and improves its reliability.

7. This system has also the advantage of being applicable not only to four-cycle but also to two-cycle engines. It therefore opens also to this kind of Diesel engines great possibilities of further development and power increase.

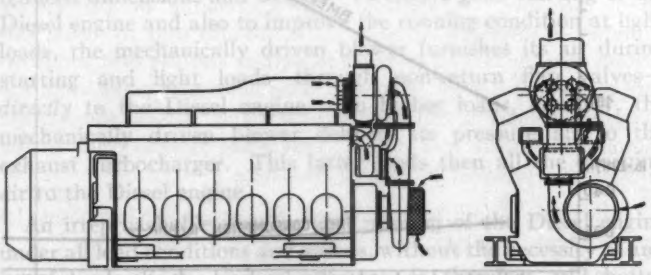


FIGURE 10.—TYPICAL ARRANGEMENT OF BUCHI-DUPLEX-TURBOCHARGING SYSTEM FOR FOUR-CYCLE TWIN-BANK DIESEL ENGINES.

Figure 10 illustrates what a "duplex" turbocharged four-cycle Diesel engine with its two charging blowers looks like.

Figure 11 shows the Entropy-and PV-diagrams of an ordinary and a "duplex"-turbocharged two-cycle Diesel engine. The thin

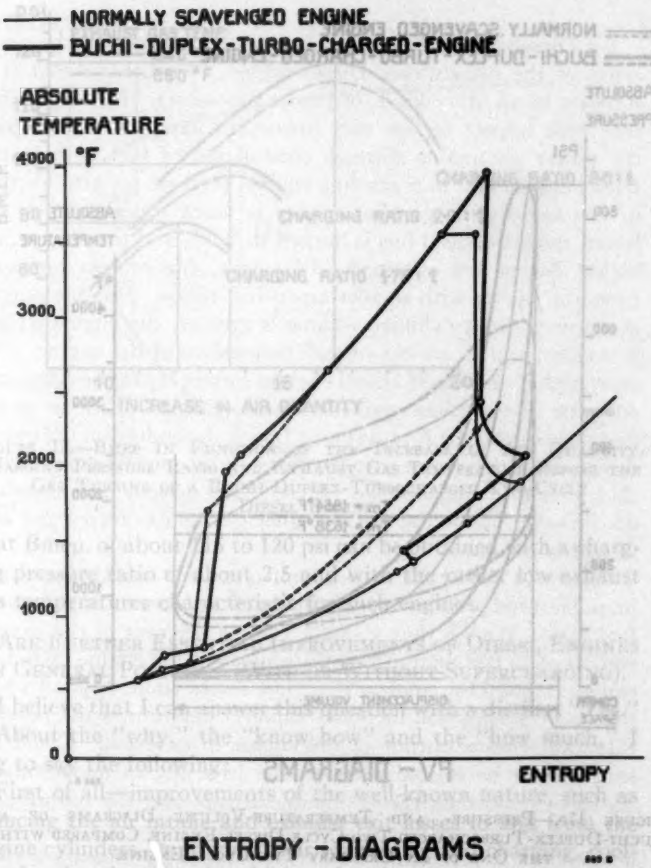


FIGURE 11A.—ENTROPY DIAGRAM OF A BUCHI-DUPLEX-TURBOCHARGED TWO-CYCLE DIESEL ENGINE, COMPARED WITH SUCH OF AN ORDINARY TWO-CYCLE ENGINE.

lines point out the cycle of the normally-scavenged engine, and the thick lines show the cycle of a "duplex" turbocharged engine.

The process temperatures are also plotted in. The mean temperature of the "duplex"-turbocharged engine is about 118°C. higher. We have made a thorough investigation and found out that due to the reduction in cylinder size—if we compare two

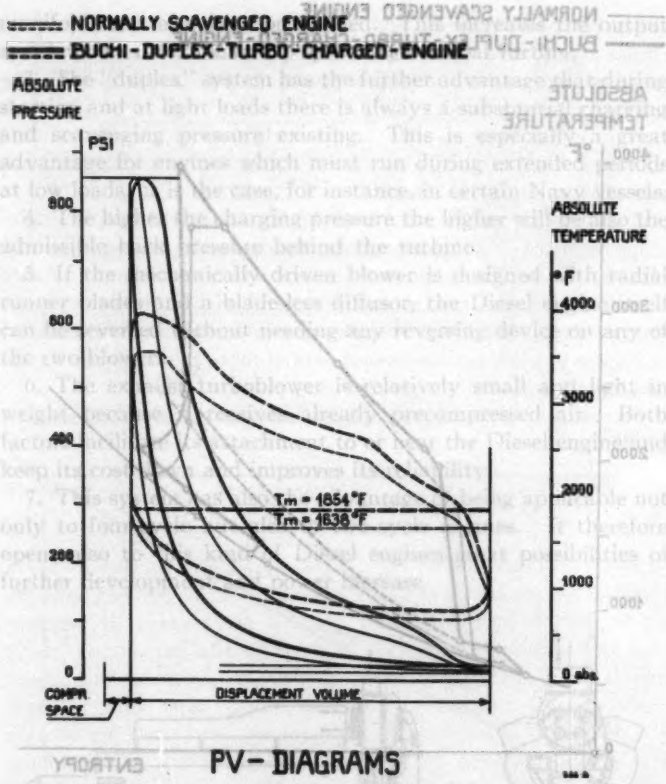


FIGURE 11B.—PRESSURE- AND TEMPERATURE-VOLUME DIAGRAMS OF A BUCHI-DUPLEX-TURBOCHARGED TWO-CYCLE DIESEL ENGINE, COMPARED WITH THE ONE OF AN ORDINARY TWO-CYCLE ENGINE.

engines of the same output—the combined stresses in the "duplex"-turbocharged engine cylinder exerted by pressure and heat transfer are rather smaller than in an ordinary two-cycle engine.

Figure 12 shows the obtainable Bmep. of a "duplex"-turbocharged two-cycle engine in function of the air weight increase compared with the normal two-cycle engine. Different charging pressure ratios and exhaust temperatures are assumed. We see

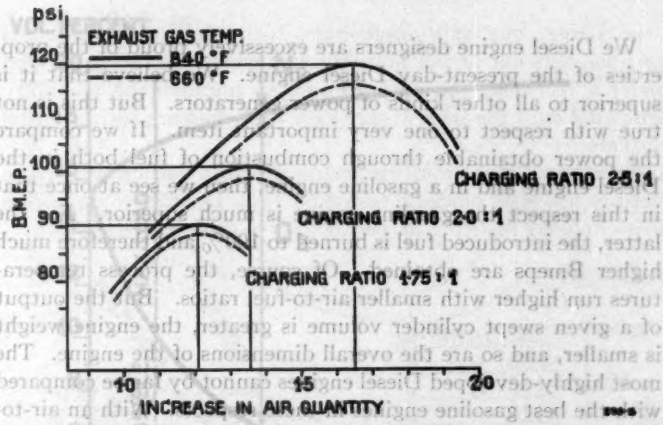


FIGURE 12.—BMEP IN FUNCTION OF THE INCREASE IN AIR QUANTITY, CHARGING PRESSURE RATIO AND EXHAUST GAS TEMPERATURE BEFORE THE GAS TURBINE OF A BUCHI-DUPLEX-TURBOCHARGED TWO-CYCLE DIESEL ENGINE.

that Bmep. of about 115 to 120 psi can be obtained with a charging pressure ratio of about 2.5 and with the rather low exhaust gas temperatures characteristic for such engines.

5. ARE FURTHER ESSENTIAL IMPROVEMENTS OF DIESEL ENGINES IN GENERAL POSSIBLE? (WITH OR WITHOUT SUPERCHARGING).

I believe that I can answer this question with a distinct "Yes."

About the "why," the "know how" and the "how much," I beg to say the following:

First of all—improvements of the well-known nature, such as reducing the air intake and exhaust exit losses to and from the engine cylinders, improving their scavenging and charging, cooling the charging air when higher charging ratios are used, etc., must first of all be considered. Often, however, a chosen engine design is not well adaptable to big improvements in these respects. Often engine designers pretend that they have achieved such improvements with relatively minor and unsubstantial changes. But also here, like everywhere else, really notable improvements can only be obtained if the chosen engine design is quite principally changed. What we need are "Milestones" in the way to still more economical, still more handsome, powerful, reliable, and less voluminous and expensive power-producers.

We Diesel engine designers are excessively proud of the properties of the present-day Diesel engine. We believe that it is superior to all other kinds of power generators. But this is not true with respect to one very important item. If we compare the power obtainable through combustion of fuel both in the Diesel engine and in a gasoline engine, then we see at once that in this respect the gasoline engine is much superior. For the latter, the introduced fuel is burned to 100% and therefore much higher Bmeps are obtained. Of course, the process temperatures run higher with smaller air-to-fuel ratios. But the output of a given swept cylinder volume is greater, the engine weight is smaller, and so are the overall dimensions of the engine. The most highly-developed Diesel engines cannot by far be compared with the best gasoline engines in these respects. With an air-to-fuel ratio of about 1.8 : 1 down to 1.5 : 1, depending on the engine design used, they come into the range of incomplete combustion, that is to say, they smoke. Present-day designs and materials do, however, allow the use of higher gas temperatures and also of higher pressures.

Figure 13 shows the smoke limit of a very well designed and manufactured present-day Diesel engine in function of the air-to-fuel ratio. We see that about 7 volume per cent of the oxygen contained in the combustion air, present in the engine cylinders, cannot be used for combustion. The smoke limit, beyond which the fuel consumption increases rapidly, determines the possible safe and reliable output of the engine in question. Could the smoke limit be shifted more to the left, i.e., to much smaller air-fuel ratios, then the admissible Bmep. and the output of a given Diesel engine could be further materially increased. This is therefore a principally sound and success-promising way to distinctly improve the Diesel engine.

We have made some development work in this direction on four-cycle engines. By tests carried through on a quite specially built engine, it could be shown that the proposed ways promise quite good results.

The following means for improving the combustion were introduced:

(*a*) A very *compact* combustion space with a small cooled surface in comparison to its volume. The compression and combustion temperature therein, especially in its center, is then a

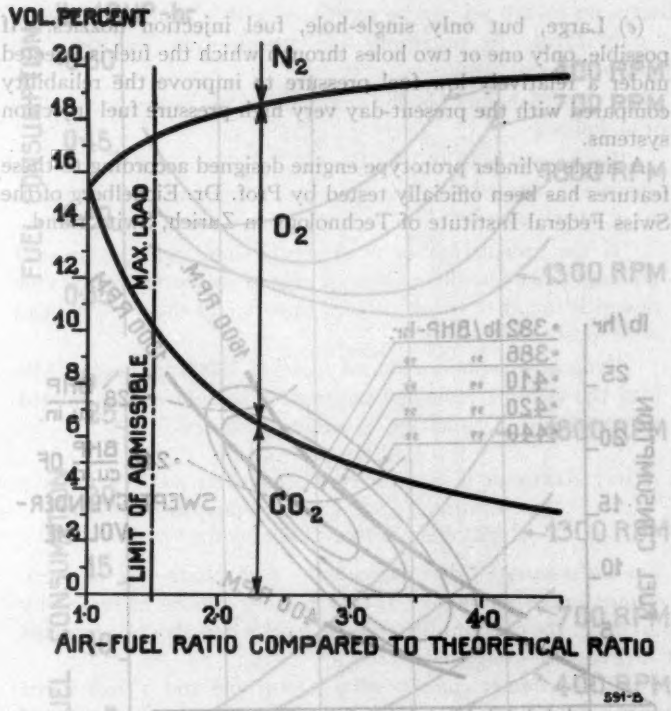


FIGURE 13.—CHEMICAL COMPOSITION OF EXHAUST GASES IN PRESENT DAY DIESEL ENGINES.

maximum. By these means not only the heat losses are diminished but also the starting and running on light loads and with low speeds is facilitated.

(*b*) The combustion space and the piston have no irregular surfaces—cuts or the like.

(*c*) Very pronounced swirl action assists the fuel distribution and its atomization and mixing with the combustion air. This is achieved in the cylinder itself and not in separated so-called precombustion chambers as in the precombustion chamber engine, the Ricardo type, etc. The use of such separated chambers materially increases heat losses and fuel consumption.

(*d*) Large inlet and exhaust valve port areas.

(*e*) Large, but only single-hole, fuel injection nozzles. If possible, only one or two holes through which the fuel is injected under a relatively low fuel pressure to improve the reliability compared with the present-day very high pressure fuel injection systems.

A single cylinder prototype engine designed according to these features has been officially tested by Prof. Dr. Eichelberg of the Swiss Federal Institute of Technology in Zurich, Switzerland.

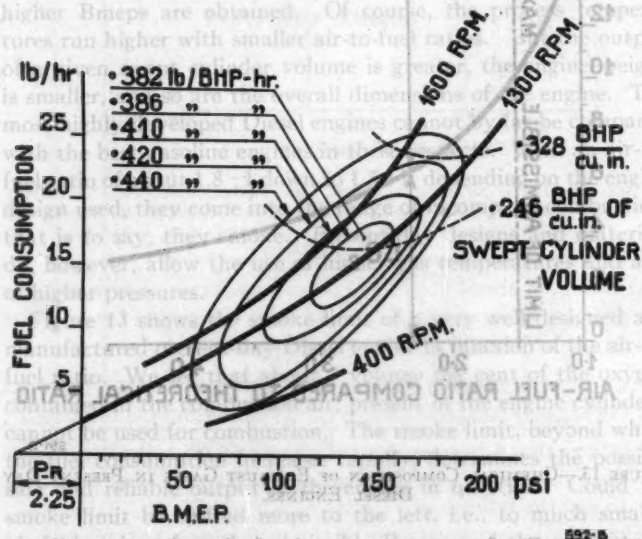


FIGURE 14.—TEST RESULTS OF A BUCHI-TELESCOPE-TYPE FOUR-CYCLE DIESEL ENGINE (SINGLE-CYLINDER PROTOTYPE WITH MECHANICALLY DRIVEN SUPERCHARGER).

Figure 14 shows the Bmep. and the fuel consumption at various engine speeds running from 400 up to 1900 Rpm. It can be seen that a Bmep. over 170 psi could be generated with a relatively low charging ratio of 1.5 : 1. Because the engine was so small a Roots blower was used as charging compressor. 15 to 20 Bhp. per liter or .246 to .328 Bhp./cu. in. swept cylinder volume is quite favorable compared with present-day Diesel engines.

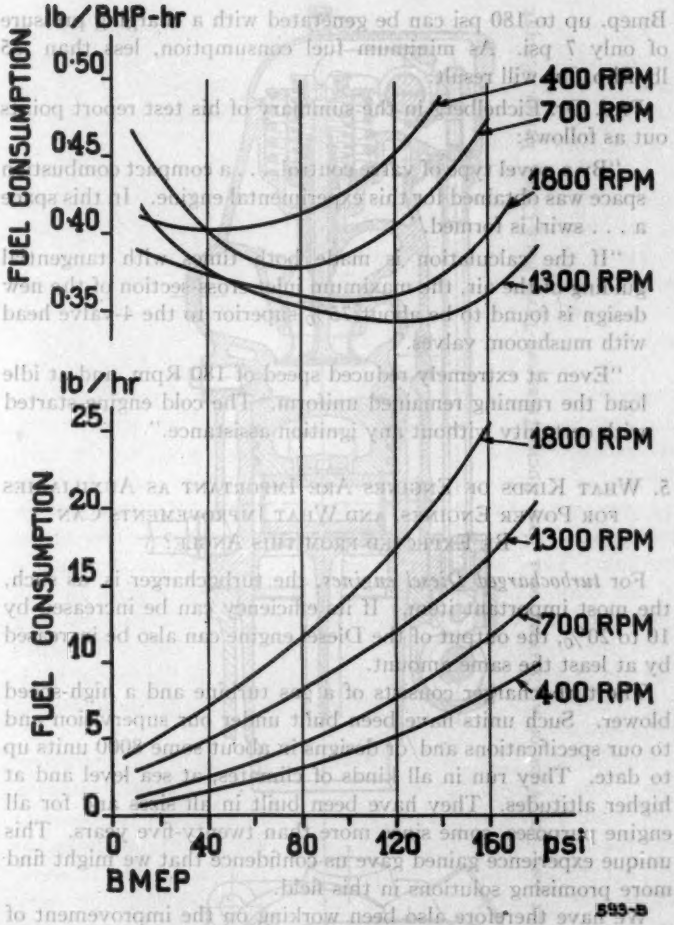


FIGURE 15.—FUEL CONSUMPTION OF A BUCHI-TELESCOPE-TYPE DIESEL ENGINE IN FUNCTION OF BMEP (MULTI-CYLINDER EXHAUST-TURBOCHARGED).

Figure 15 illustrates the Bmep. and the fuel consumption per metric Bhp./hr. when the system is applied to a multicylinder engine working with turbocharging and having the same cylinder dimensions as the tested engine. We see that at 1300 Rpm.

Bmep. up to 180 psi can be generated with a charging pressure of only 7 psi. As minimum fuel consumption, less than .35 lb/Bhp./hr. will result.

Prof. Dr. Eichelberg in the summary of his test report points out as follows:

"By a novel type of valve control, . . . a compact combustion space was obtained for this experimental engine. In this space a . . . swirl is formed."

"If the calculation is made both times with tangential guiding of the air, the maximum inlet cross-section of the new design is found to be about 75% superior to the 4-valve head with mushroom valves."

"Even at extremely reduced speed of 180 Rpm. and at idle load the running remained uniform. The cold engine started with certainty without any ignition assistance."

5. What Kinds of Engines Are Important as Auxiliaries for Power Engines, and What Improvements Can Be Expected from this Angle?

For *turbocharged Diesel engines*, the turbocharger is, as such, the most important item. If its efficiency can be increased by 10 to 20%, the output of the Diesel engine can also be increased by at least the same amount.

The turbocharger consists of a gas turbine and a high-speed blower. Such units have been built under our supervision and to our specifications and/or designs in about some 8000 units up to date. They run in all kinds of climates, at sea level and at higher altitudes. They have been built in all sizes and for all engine purposes, some since more than twenty-five years. This unique experience gained gave us confidence that we might find more promising solutions in this field.

We have therefore also been working on the improvement of both the blower and the turbine.

Figure 16 shows a vertical section and a top view of a cylinder of a new Buchi-Telescope-Type four-cycle Diesel engine which includes the features mentioned in this chapter.

Figure 17 shows a section through a turbocharger for four-cycle Diesel engines built by the Elliott Company, Jeannette, Pennsylvania. All its vital parts can be clearly seen.

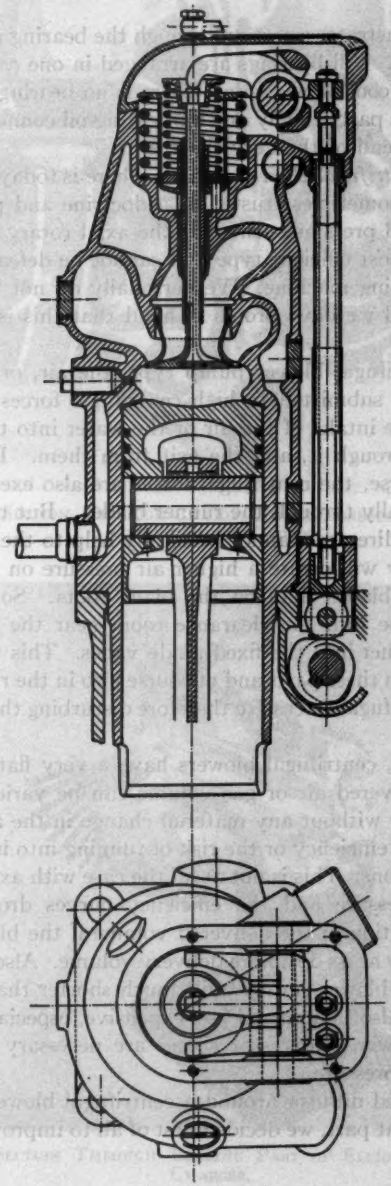


FIGURE 16.—VERTICAL SECTION AND TOP VIEW OF A CYLINDER OF A NEW BUCHI-TELESCOPE-TYPE FOUR-CYCLE DIESEL ENGINE.

Figure 18 illustrates a section through the bearing part of such a turbocharger. All bearings are arranged in one casing, which is fixed on the cool blower side. There is no bearing on or near the hot turbine parts. Only one lubricating oil connection exists on the blower end of the very short unit.

(*a*) *New Centrifugal Blower Design.* There is today a big widespread belief, sometimes trusted as a doctrine and pointed out in the technical press as such, that the axial rotary compressor design is the most efficient type and cannot be defeated by any other compressing machine. We personally do not believe this hypothesis, and we have proofs in hand that this is effectively not so.

In the centrifugal blower pump type, the air, or the water, respectively, is submitted to high centrifugal forces. This certainly helps the intake of the air or the water into the runners, the passage through it, and the exit from them. In the axial blower, of course, the centrifugal forces are also exerted on the air passing axially through the runner blades. But these forces, being radially directed, cannot be of any help to the flow of the medium. They will create a higher air pressure on the outside of the blower blades than on the blade roots. Some air will therefore escape into the clearance room near the blades and from there further into the fixed guide vanes. This will disturb the flow of air in these parts and of course also in the runner blading. The centrifugal forces are therefore disturbing the air flow in axial blowers.

Furthermore, centrifugal blowers have a very flat characteristic. The delivered air or gas volume can be varied within a very wide limit without any material change in the air pressure and the blower efficiency or the risk of running into inadmissible surging conditions. This is not at all the case with axial blowers. There, the pressure and the efficiency curves drop off very quickly in function of the delivered volume if the blower is not working exactly at its optimum delivery volume. Also physically the centrifugal blower can be built much shorter than the axial blower. It is also lighter and less expensive, especially because many times fewer compressor stages are necessary to get the same delivery pressure.

Since the fixed diffusor around a centrifugal blower runner is the least efficient part, we decided first of all to improve this part

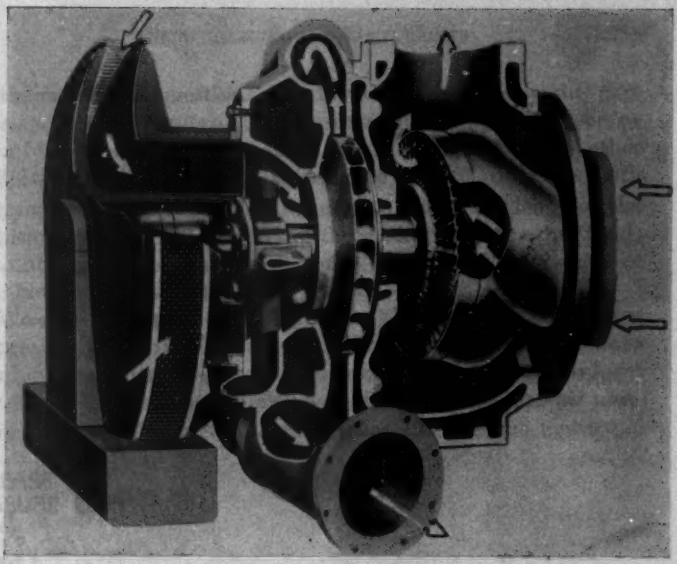
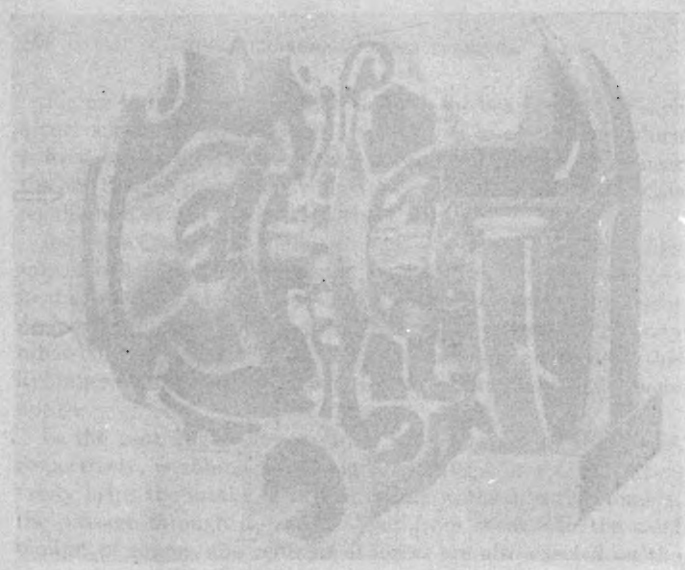
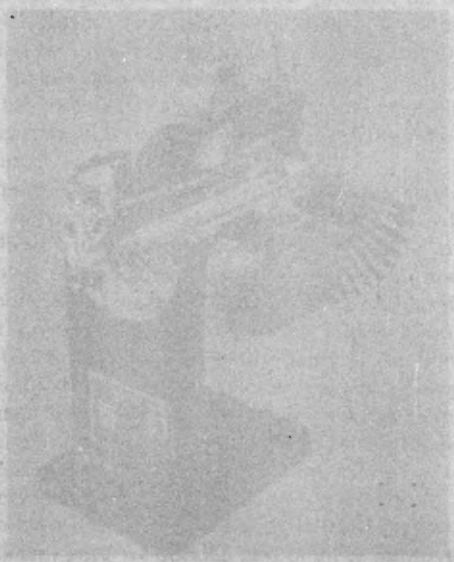


FIGURE 17.—SECTION THROUGH ELLIOTT-BUCHI-TURBOCHARGER FOR FOUR-CYCLE DIESEL ENGINES.

FIGURE 18.—SECTION THROUGH BEARING PART OF ELLIOTT-BUCHI-TURBO-CHARGER.





of the machine in question. It is known that up to now this part was executed either as a spiral, or as a radially open diffusor, or as a bladed one, etc. It is surrounded by a spiral or any other collecting casing. Carefully designed bladed diffusors with rectangular through sections have a maximum efficiency of the transformation of the velocity of the medium (air or other fluid) into pressure of only about 40 to 65%, depending on the size of the blower and the diffusor type used.

We use now, in contradistinction to the conventional diffusor types, only a restricted number of diffusors with *circular* sections per blower runner. Diffusors with circular through sections have far higher efficiencies of conversion of air velocity into air pressure. We see this clearly from Figure 19. In this figure the

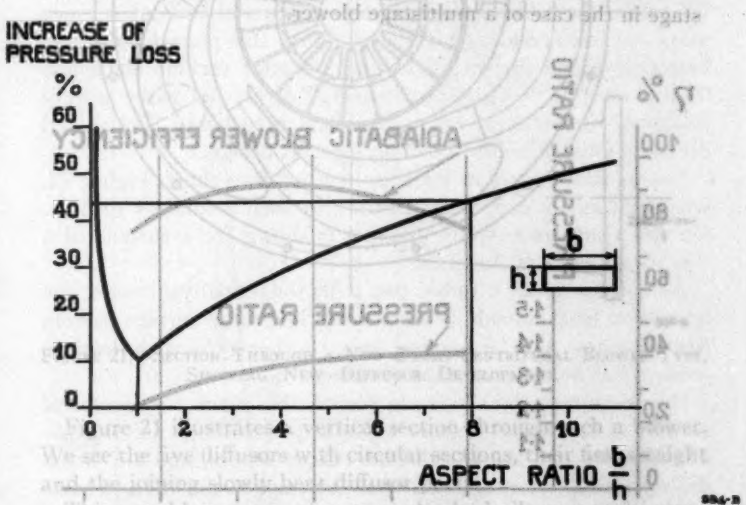


FIGURE 19.—INCREASE OF LOSSES IN A STRAIGHT DIFFUSOR WITH RECTANGULAR, COMPARED WITH ONE OF CIRCULAR THROUGH-FLOW SECTION.

increase of pressure losses in per cent in straight diffusors with square and rectangular section is shown compared with the pressure losses in straight diffusors with circular section. The pressure loss in a square-sectioned diffusor is 10% and, in a rectangular diffusor with an aspect ratio of 2 : 1, even 17.5% higher than in a diffusor with circular section. Our diffusors are

now in the first part, where the overwhelming percentage of the transformation of velocity into pressure takes place, built with straight axis and with the most favorable opening angle. We have therefore already secured the main item for a good efficiency. In order to keep the efficiency also in the second, adjoining parts on a high level, only slowly-bent diffusor parts, but also of gradually increasing circular through section, are used. They convert the velocity further efficiently into pressure and direct with a minimum of losses the air into the surrounding casing. The second parts of the circular diffusors are bent to reduce the outside diameter of the blower to a minimum. In the casing the velocity is kept low so as to spare further losses. The air leaves the casing directly or via another straight diffusor, or to the next stage in the case of a multistage blower.

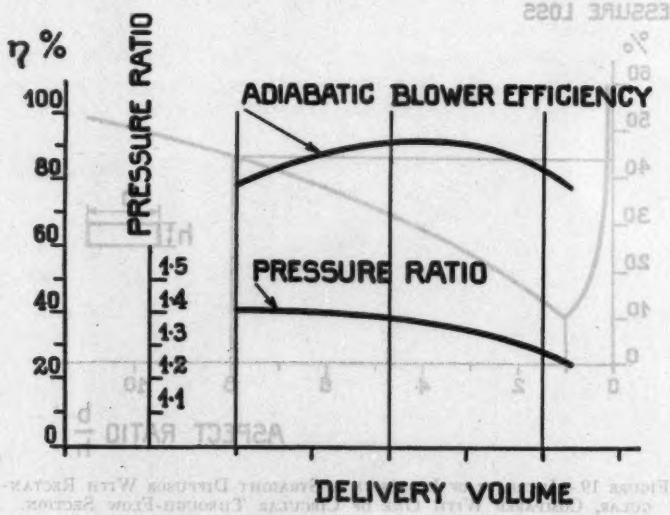


FIGURE 20.—PRESSURE RATIO AND ADIABATIC EFFICIENCY OF NEW TYPE CENTRIFUGAL BLOWER TYPE.

The diagram of tests with such a new blower, Figure 20, shows a maximum adiabatic efficiency of 92%. We see also that this high efficiency keeps quite high within a very wide range of the

delivered air volume. This high efficiency is also obtained with a higher pressure ratio (about 1.4 : 1) than hitherto at the same circumferential speed.

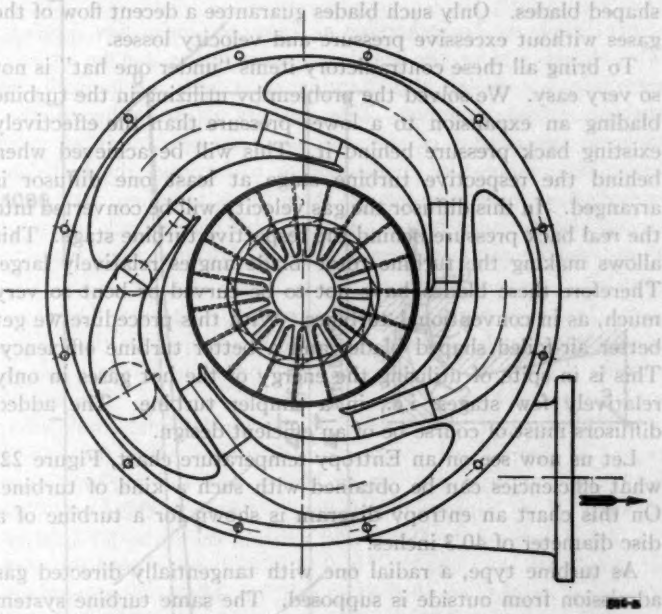


FIGURE 21.—SECTION THROUGH A NEW BUCHI-CENTRIFUGAL BLOWER TYPE, SHOWING NEW DIFFUSOR DEVELOPMENT.

Figure 21 illustrates a vertical section through such a blower. We see the five diffusors with circular sections, their first straight and the joining slowly bent diffusor parts.

This new blower can, of course, also be built as a multi-stage machine, as is necessary for gas turbines, turbojet engines and industrial purposes.

(*b*) *Improvement of the Gas Turbine Design Itself.* A gas turbine which has to work with gases of very high temperature should be very simple and should have a minimum of turbine stages, turbine discs, guiding apparatus, and few costly blades. Every item must be of robust design and of heat-resisting material. But on the contrary, we know that the higher the pressure ratio

in one turbine stage is, the lower the efficiency will be, also with this kind of engine—similarly as with blowers. High pressure ratios per stage make it also more difficult to create good airfoil-shaped blades. Only such blades guarantee a decent flow of the gases without excessive pressure and velocity losses.

To bring all these contradictory items "under one hat" is not so very easy. We solved the problem by utilizing in the turbine blading an expansion to a lower pressure than the effectively existing back pressure behind it. This will be achieved when behind the respective turbine stage at least one diffusor is arranged. In this diffusor the gas velocity will be converted into the real back pressure behind the respective turbine stage. This allows making the turbine rotor blade angles relatively large. Therefore these blades have not to be curved or bent so very much, as in conventional turbines. With this procedure we get better air-foiled shaped blades and a better turbine efficiency. This is in spite of utilizing the energy of the hot gases in only relatively few stages, i.e., in a simpler turbine. The added diffusors must of course be of an efficient design.

Let us now see on an Entropy-temperature chart, Figure 22, what efficiencies can be obtained with such a kind of turbine. On this chart an entropy diagram is shown for a turbine of a disc diameter of 40.3 inches.

As turbine type, a radial one with tangentially directed gas admission from outside is supposed. The same turbine system can of course also be applied to any other type with axial or axial-radial flow direction of the gases.

On the upper right side we see on the chart a schematical section of the radial-turbine type in question. O are the gas inlet nozzles, 1-2 the flat turbine runner blades, and 2-3 is the diffusor. The latter is, in this case, a straight one with a circular section.

We see that with a relatively large turbine wheel, which corresponds to the size used in gas turbines and turbo jets, an adiabatic efficiency of 93.2% will be obtained. With smaller turbine runners for instance of 10 in. diameter still 88.6 per cent adiabatic efficiency can be produced. Such smaller sizes are needed especially for turbocharging internal combustion engines.

Turbine experts will surely agree that these are exceptionally good figures for a pressure ratio of the fluid of about 1.4 : 1.

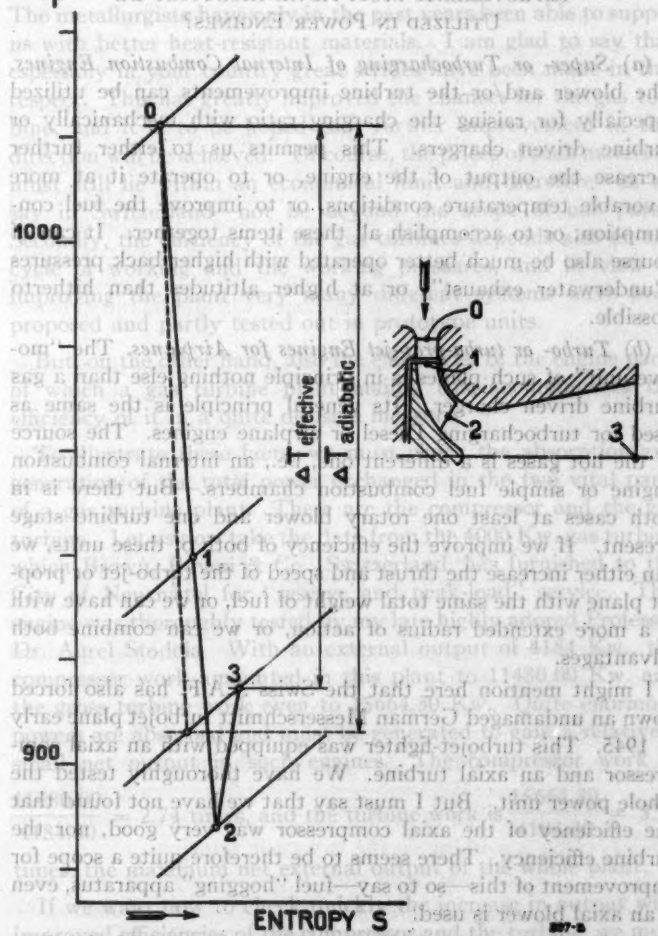


FIGURE 22.—ENTROPY DIAGRAM OF A NEW BUCHI-RADIAL TYPE GAS TURBINE WITH DIFFUSOR.

6. Where Can the Suggested Blower and/or the Turbine Improvements Most Advantageously Be Utilized in Power Engines?

(*a*) *Super- or Turbocharging of Internal Combustion Engines.* The blower and/or the turbine improvements can be utilized especially for raising the charging ratio with mechanically or turbine driven chargers. This permits us to either further increase the output of the engine, or to operate it at more favorable temperature conditions, or to improve the fuel consumption, or to accomplish all these items together. It can of course also be much better operated with higher back pressures ("underwater exhaust") or at higher altitudes than hitherto possible.

(*b*) *Turbo- or turbo-prop-jet Engines for Airplanes.* The "motive-unit" of such planes is in principle nothing else than a gas turbine driven charger. Its general principle is the same as used for turbocharging Diesel or airplane engines. The source of the hot gases is a different one, i.e., an internal combustion engine or simple fuel combustion chambers.—But there is in both cases at least one rotary blower and one turbine stage present. If we improve the efficiency of both of these units, we can either increase the thrust and speed of the turbo-jet or prop-jet plane with the same total weight of fuel, or we can have with it a more extended radius of action, or we can combine both advantages.

I might mention here that the Swiss A.A.F. has also forced down an undamaged German Messerschmitt turbojet plane early in 1945. This turbojet-fighter was equipped with an axial compressor and an axial turbine. We have thoroughly tested the whole power unit. But I must say that we have not found that the efficiency of the axial compressor was very good, nor the turbine efficiency. There seems to be therefore quite a scope for improvement of this—so to say—fuel "hogging" apparatus, even if an axial blower is used.

(*c*) *Gas Turbines for Stationary, Locomotive, and Marine Purposes.* In such engines, combustible gases, liquids, or solid fuels are—after combustion in a combuster—directly admitted into turbines as driving means instead of steam, as with steam turbines. "Combustion Gas Turbines" would therefore be a more

correct title. Its rather low thermal efficiency depends mainly on a high initial temperature of the introduced driving gases. The metallurgists have only in the past years been able to supply us with better heat-resistant materials. I am glad to say that especially in your country great strides have been made in this respect. This has greatly improved the chances for the gas turbine, and it is to be hoped that further improvement in this direction will be achieved. Of course, the prices of such materials must still lie within an economical limit and therefore—as we say in Switzerland—not be beyond the scope of our purse. Secondly, the efficiency of the gas turbines depends also on the cycle of working and the working pressures, and in view of improving the plant very many different systems have been proposed and partly tested out in prototype units.

But, on the other hand, also the efficiencies of the engine parts of which a gas turbine plant mainly consists influences the efficiency of it in a quite decisive measure.

To illustrate these facts, we must check the absorption and generation of the total power exchanged in the two vital parts of a gas turbine plant. These are the compressor and the gas turbine. Let us now take the data from the 4000 Kw. gas turbine which Brown, Boveri & Co., Switzerland, has furnished to the City of Neuchâtel for "reserve and peak-load" service. This engine was thoroughly tested by my late highly adored Professor Dr. Aurel Stodola. With an external output of 4184 Kw., the compressor work amounted in this plant to 11480.00 Kw. and the gross turbine work even to 15664.30 Kw. Quite enormous powers are absorbed and must be generated to gain a relatively small net output in such engines. The compressor work is $\frac{11480.00}{4184.30} = 2.74$ times, and the turbine work is $\frac{15664.30}{4184.30} = 3.74$ times, the maximum net external output of the whole plant.

If we want now to check quickly the increase in output with improved efficiencies of the compressor and the turbine, we must keep in mind that such improvements, for instance in per cent, amount to 2.74 and 3.74 times respectively more, counted on the net output of the gas turbine alone than its percentage on the compressor or turbine work in itself.

5% less work on the compressor means therefore $5 \times 2.74 = 13.70\%$ more external power on the turbine coupling.

4% more gross turbine work means $4 \times 3.74 = 14.97\%$ more external power at the turbine coupling.

The *total increase in output* due to the two assumed improvements of the compressor and the turbine is therefore *28.67%*. This increase in output is, due to the large negative work exchanged within such a plant itself, quite considerable. The saving in fuel consumption will be a little smaller, because the combustion air to be heated to a certain maximum temperature in the combuster does not get so warm in a better compressor as it would in a less efficient one. But the thermal efficiency will still be raised in our example by about 19.3%.

If the compared gas turbine plants would work with exhaust gas preheaters for the combustible air, which they do not, all the outputs and the efficiencies would be of course bigger. But the relative figures compared would still be in the same proportion in favor of the plant using the more efficient compressor and turbine.

7. Gas Turbine with a Diesel Engine Acting as Its Gas Producer.

In this last chapter of my lecture, I want to speak a few words about still another power plant system, the principle of which I have suggested quite a time ago, and which uses high pressure charging. See, for instance, the congratulatory book which the former students and friends of Prof. Dr. Aurel Stodola, Zurich, have presented to him on the occasion of his seventieth anniversary. Sulzer Bros., Winterthur, have published lately some test results of a prototype of such a kind of prime mover, where the Diesel engine is working on the two-cycle principle.

Let us now see how, with such a prime mover, a four-cycle Diesel engine will act as a gas producer for the gas turbine. The Diesel engine drives then only the necessary air charging and scavenging compressors.

Figure 23 shows the PV-diagram of such a Diesel engine. As can be seen from the suction stroke of the engine, a not too high, constant charging pressure of 28.4 psi Gauge is here assumed.

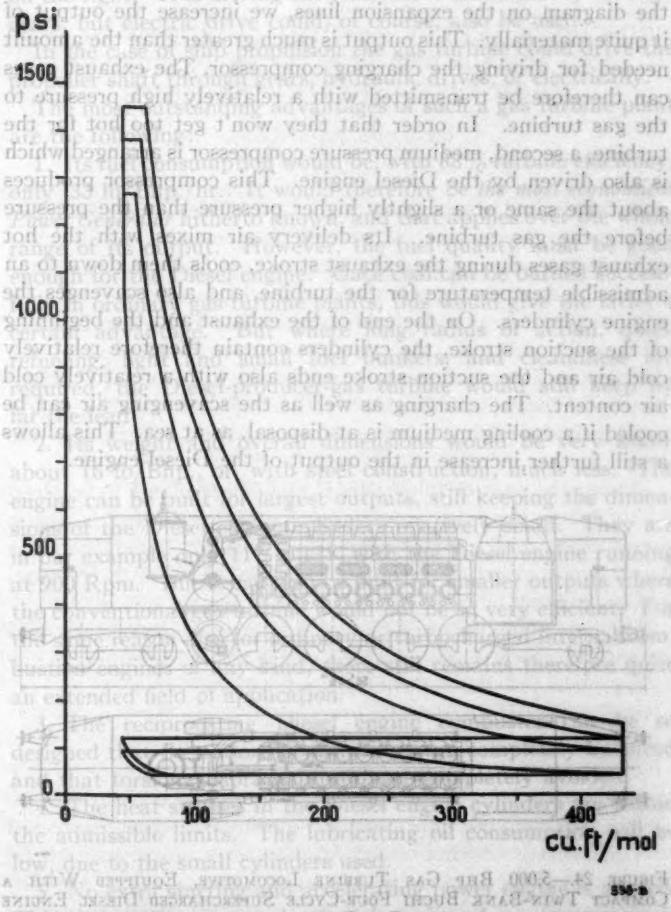


FIGURE 23.—PV-DIAGRAMS OF A HIGHLY SUPERCHARGED AND SCAVENGED FOUR-CYCLE DIESEL ENGINE, WORKING AS A GAS PRODUCER FOR AN EXTERNALLY POWER GENERATING GAS TURBINE.

If we introduce more and more fuel into the Diesel engine, as is shown by the air-to-fuel ratios $\lambda = 3.2$ to 1.5, as marked in the diagram on the expansion lines, we increase the output of it quite materially. This output is much greater than the amount needed for driving the charging compressor. The exhaust gases can therefore be transmitted with a relatively high pressure to the gas turbine. In order that they won't get too hot for the turbine, a second, medium pressure compressor is arranged which is also driven by the Diesel engine. This compressor produces about the same or a slightly higher pressure than the pressure before the gas turbine. Its delivery air mixes with the hot exhaust gases during the exhaust stroke, cools them down to an admissible temperature for the turbine, and also scavenges the engine cylinders. On the end of the exhaust and the beginning of the suction stroke, the cylinders contain therefore relatively cold air and the suction stroke ends also with a relatively cold air content. The charging as well as the scavenging air can be cooled if a cooling medium is at disposal, as at sea. This allows a still further increase in the output of the Diesel engine.

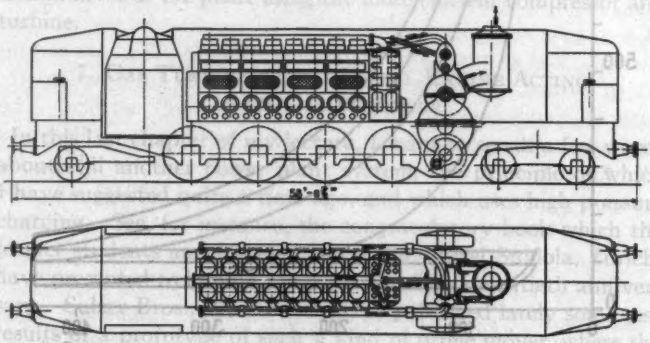


FIGURE 24.—5,000 BHP GAS TURBINE LOCOMOTIVE, EQUIPPED WITH A COMPACT TWIN-BANK BUCHI FOUR-CYCLE SUPERCHARGED DIESEL ENGINE GAS PRODUCER.

To give a rough idea about the necessary dimensions, Figure 24 shows the installation of such a power plant of 5000 Bhp. output on a locomotive. The crossheads of the engine cylinders are arranged as charging compressors and the double-acting tandem

scavenging pump is on the engine end. The gas turbine is shown on the right. It is driving the locomotive mechanically through gears, but electric drive could, of course, also be used.

In the case of ship propulsion the gas turbine could drive the propeller shaft through gears, hydraulic drives, or electrically.

The most outstanding advantages of such a gas turbine plant are the following:

1. Its fuel consumption would be, with 88% turbine efficiency, only .33 lb/Bhp./hr. It would therefore be *the most economical Power Generator* hitherto known, and that applies over the whole range of its output. However, the fuel quality must be good enough for the Diesel engine. Once coal can be burned successfully in ordinary gas turbine plants, this would give the latter a certain advantage. But where long radius of action, quick refueling, light and small fuel bunkers, and cleanliness are required, the Diesel-producer-gas turbine would still keep by far the lead!

2. Its weight and overall dimensions would be very small, about 16 lb/Bhp., or, with steel construction, much less. The engine can be built for largest outputs, still keeping the dimensions of the Diesel engine cylinders relatively small. They are in our example only 11″ × 12″, with the Diesel engine running at 900 Rpm. But it can also be built for smaller outputs where the conventional gas turbine would not be so very efficient. For the same reason also for ordinary or turbocharged internal combustion engines of any kind, there still remains therefore quite an extended field of application.

3. The reciprocating Diesel engine combuster can be so designed that the actions of the masses are completely balanced and that torsional vibrations, etc., are completely avoided.

4. The heat stresses in the Diesel engine cylinders are within the admissible limits. The lubricating oil consumption will be low, due to the small cylinders used.

5. No extra starting and accelerating power engine is needed. This is very important when quick action of a prime mover is needed. Such a necessity is a very undesirable addition with an ordinary gas turbine plant.

6. The gas turbine can work with high back pressures because the gases are forced out by the positively displacing piston gas producer.

PRINCIPLES UNDERLYING THE CHOICE OF A CYCLE AND MACHINERY FOR STEAM PROPELLED HIGH SPEED SURFACE WARSHIPS.

By Rudolph Michel.

If the desired characteristics of a particular type of surface warship are specified, it is possible to determine within close limits the steam cycle and machinery types best suited to make use of the most recent advances in the engineering sciences, so that a more efficient and reliable power plant than any previously built, will result. In the central power station industry, the incentive for improvement is greater profit. In the case of Naval vessels the incentive is increased reliability under fighting conditions, greater cruising radius and better fighting qualities of the ship.

During the past century, Naval and Shore plants have alternately taken the lead in inaugurating improvements in steam power plant design. Recurrent wars make it imperative that Naval power plant designers be the equal of those to be found in any industry.

The following characteristics of the vessel, at full power and at cruising speed, have a direct bearing on the selection of the steam cycle and the types of propulsion machinery.

(*a*) Propeller Rpm.

(*b*) Radius of action desired.

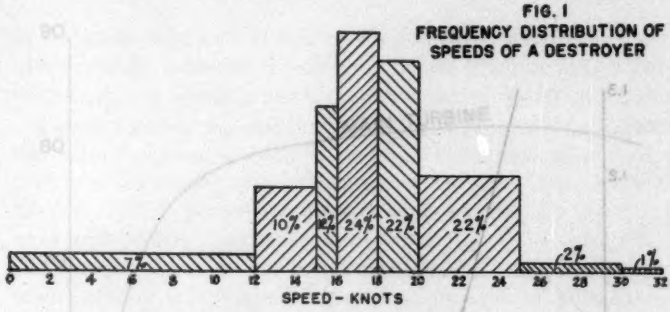
(*c*) Shaft horsepower.

(*d*) Total time to be spent at full power and at cruising speeds.

Figure 1 shows the time underway at different speeds expressed as a percentage of total operating time for a destroyer. About 80 per cent of the time is spent at cruising speeds, which are considered to be 15 to 25 knots, with only about 1 per cent of operating time at full power. We will therefore assume that the largest possible cruising radius be established at 20 knots speeds.

After the lines of the ship's hull have been determined, a model test is made to predict the relations between ship's speed, shaft horsepower and propeller Rpm. and efficiency.

FIG. I
FREQUENCY DISTRIBUTION OF SPEEDS OF A DESTROYER



Cruising radius is defined by the following relation:

$$C.\,R. = \frac{C - D}{A \times B} \times S \qquad (1)$$

where

C. R. = cruising radius, in nautical miles.
A = Shp.
B = fuel rate, lbs. oil burned per Shp.-hr.
S = speed in nautical miles per hour.
C-D = fuel oil stowage, lbs.
C = total displacement of vessel, lbs.
D = machinery plus hull weight, lbs.

The displacement of the vessel is fixed by persons other than the power plant designer. It is therefore obvious that in order to make the cruising radius at any speed a maximum, A × B must be a minimum at that speed and D a minimum.

Figure 2 shows characteristic curves for a typical high speed surface warship, in this case a destroyer. The hull resistance determines the magnitude of ordinate A at any speed, while the efficiency of the propulsion machinery determines the value of ordinate B. Since this paper is not concerned with hull resistance, our problem is to examine ways of reducing B and D in formula (1).

The propulsion machinery may be designed to give various shapes to the economy curve of Figure 2. For example, the power plant could be designed so that best economy is obtained at F.P., as indicated by the dotted curve. Since it is desired to obtain the maximum possible cruising radius at 20 knots, it is desirable to design the propulsion machinery so that the ordinate

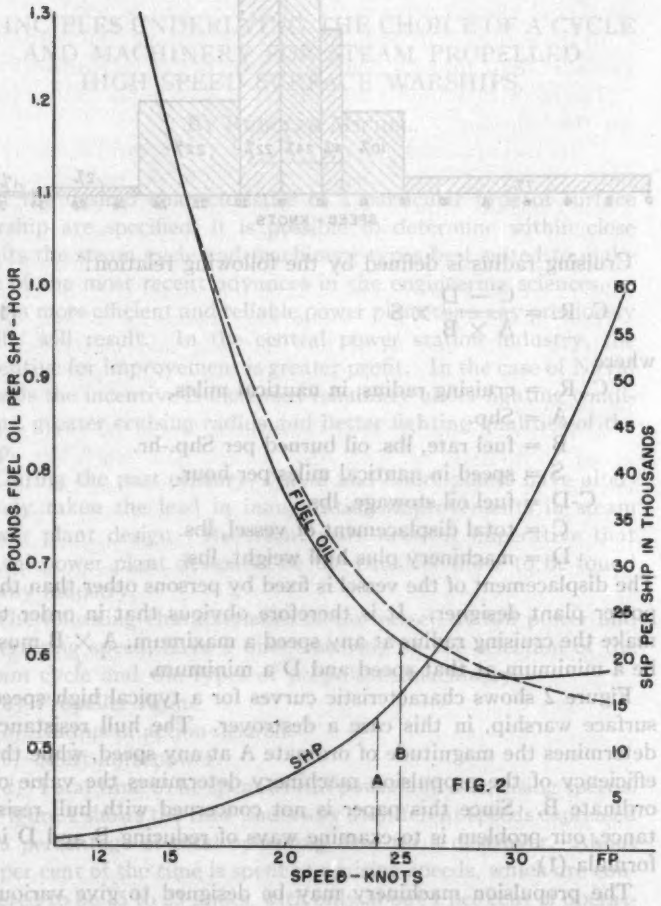

1.3
1.2
1.1
1.0
0.9
0.8
0.7
0.6
0.5
POUNDS FUEL OIL PER SHP - HOUR
60
55
50
45
40
35
30
25
20
15
10
5
SHP PER SHIP IN THOUSANDS
FUEL OIL
SHP
A
B
12
15
20
25
30
FP
SPEED - KNOTS

FIG. 2

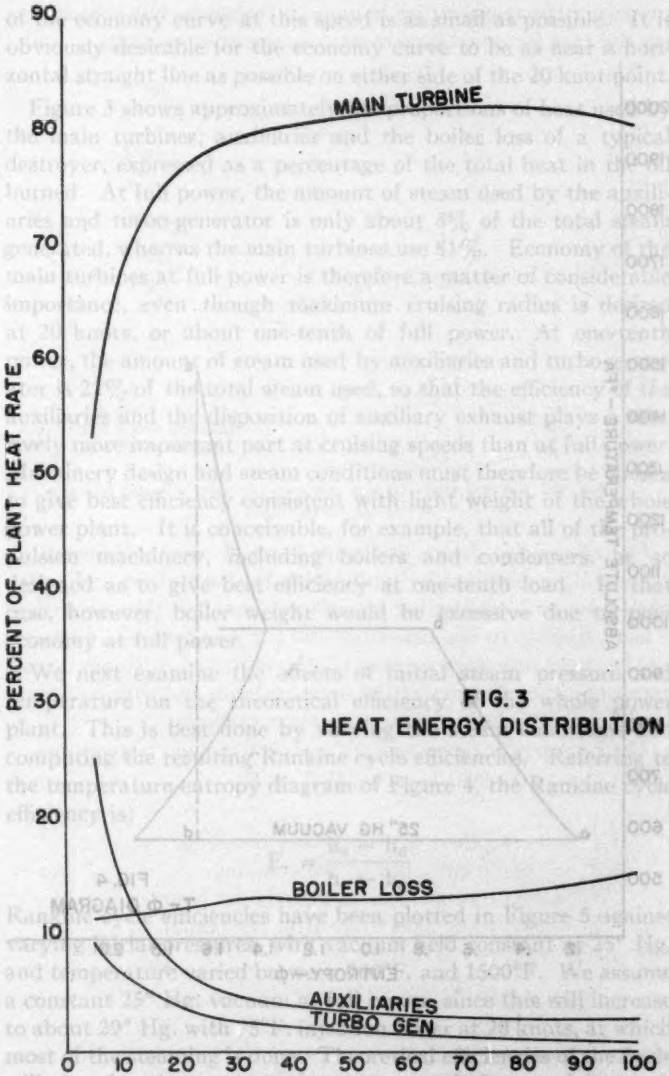


FIG. 3
HEAT ENERGY DISTRIBUTION

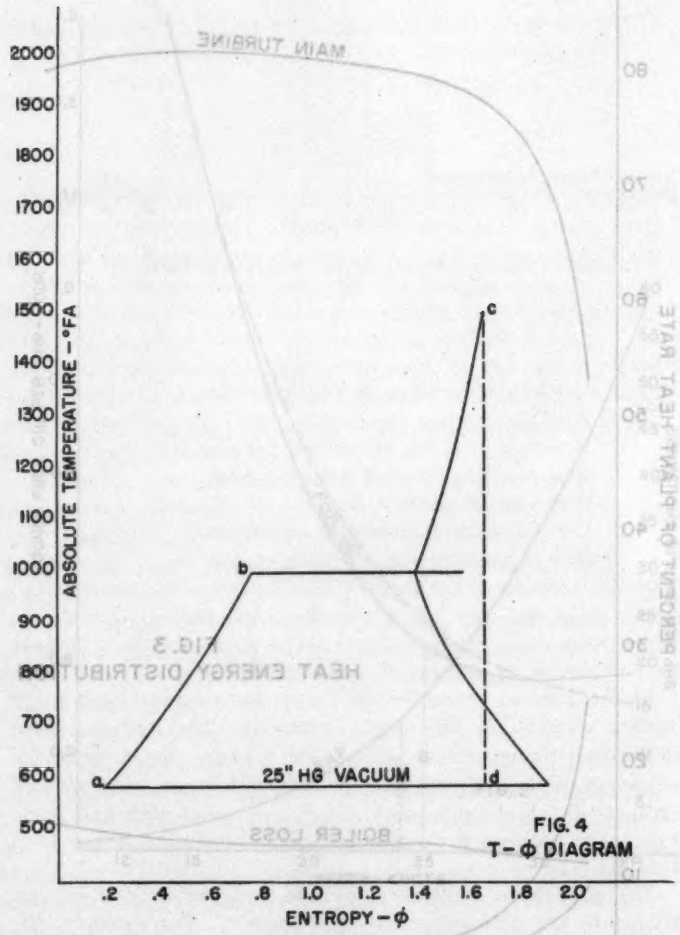


FIG. 4
T — ϕ DIAGRAM

of the economy curve at this speed is as small as possible. It is obviously desirable for the economy curve to be as near a horizontal straight line as possible on either side of the 20 knot point.

Figure 3 shows approximately the proportions of heat used by the main turbines, auxiliaries and the boiler loss of a typical destroyer, expressed as a percentage of the total heat in the oil burned. At full power, the amount of steam used by the auxiliaries and turbo-generator is only about 3% of the total steam generated, whereas the main turbines use 81%. Economy of the main turbines at full power is therefore a matter of considerable importance, even though maximum cruising radius is desired at 20 knots, or about one-tenth of full power. At one-tenth power, the amount of steam used by auxiliaries and turbo-generator is 22% of the total steam used, so that the efficiency of the auxiliaries and the disposition of auxiliary exhaust plays a relatively more important part at cruising speeds than at full power. Machinery design and steam conditions must therefore be chosen to give best efficiency consistent with light weight of the whole power plant. It is conceivable, for example, that all of the propulsion machinery, including boilers and condensers, be so designed as to give best efficiency at one-tenth load. In that case, however, boiler weight would be excessive due to poor economy at full power.

We next examine the effects of initial steam pressure and temperature on the theoretical efficiency of the whole power plant. This is best done by varying the steam conditions and computing the resulting Rankine cycle efficiencies. Referring to the temperature-entropy diagram of Figure 4, the Rankine cycle efficiency is:

$$E_r = \frac{h_c - h_d}{h_c - h_a}$$

Rankine cycle efficiencies have been plotted in Figure 5 against varying initial pressures, with vacuum held constant at 25″ Hg. and temperature varied between 900°F. and 1500°F. We assume a constant 25″ Hg. vacuum at full power, since this will increase to about 29″ Hg. with 75°F. injection water at 20 knots, at which most of the steaming is done. Theoretical efficiencies of the cycle will therefore be greater at cruising speed than at full speed. We note that approximately equal gains in efficiency result from

increasing the temperature in equal increments, but that this is not the case for equal incremental increases in pressure.

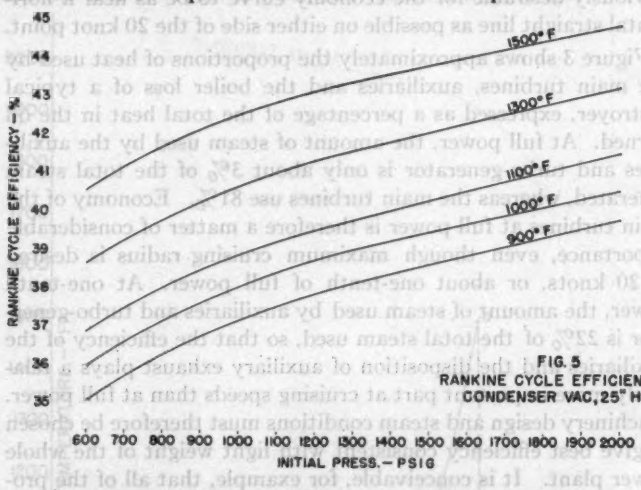


FIG. 5
RANKINE CYCLE EFFICIENCY
CONDENSER VAC, 25" HG

The efficiencies shown in Figure 5 are of course decreased in the actual power plant by multiplying by the turbine and boiler efficiencies and subtracting an amount equal to the percentage of heat supplied to the auxiliaries and turbo-generator. It therefore becomes necessary to examine the influence of steam conditions upon the efficiency of the main turbines and turbine driven auxiliaries. We first consider the main turbines.

The horsepower developed by a steam turbine is given by,

$$H.P. = 0.000393\ (h_1 - h_2)\ E.E. \times W \qquad (2)$$

where,

H.P. = horsepower at the turbine coupling.
E.E. = engine efficiency at the coupling.
W. = steam consumption, pounds per hour.
$h_1 - h_2$ = isentropic heat drop, Btu. per pound.

The engine efficiency is defined by,

$$E.E. = \frac{2545}{S.R.\ (h_1 - h_2)} \qquad (3)$$

where,

S.R. = steam rate, pounds per Shp. per hour.
$h_1 - h_2$ = isentropic heat drop, Btu. per pound.

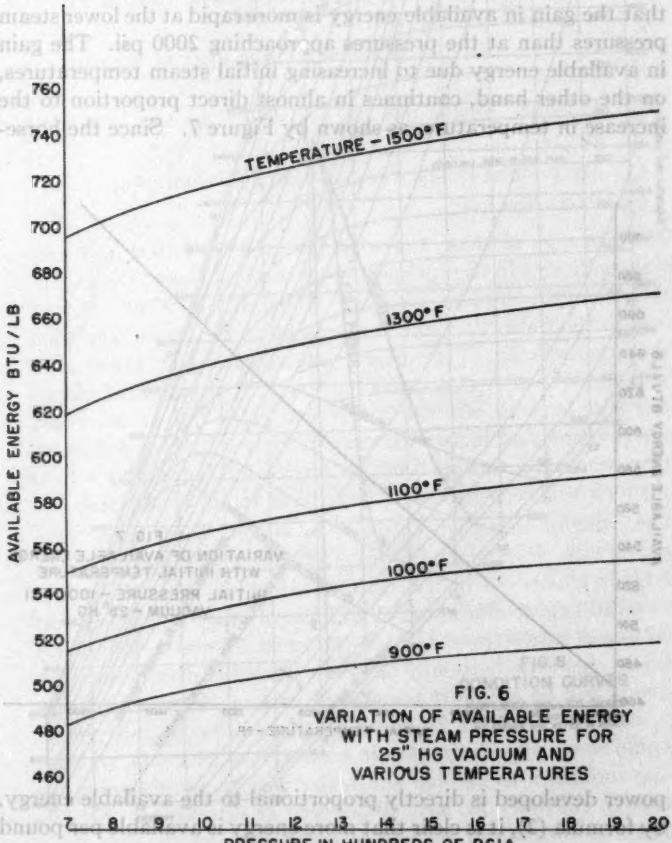


FIG. 6
VARIATION OF AVAILABLE ENERGY WITH STEAM PRESSURE FOR 25" HG VACUUM AND VARIOUS TEMPERATURES

The isentropic heat drop, or energy available to perform work in the turbine, increases with the initial pressure and temperature and also with an increase in vacuum. The rate of increase in available energy with pressure, holding vacuum constant, is shown in Figure 6, plotted for various temperatures. It is seen that the gain in available energy is more rapid at the lower steam pressures than at the pressures approaching 2000 psi. The gain in available energy due to increasing initial steam temperatures, on the other hand, continues in almost direct proportion to the increase in temperature, as shown by Figure 7. Since the horse-

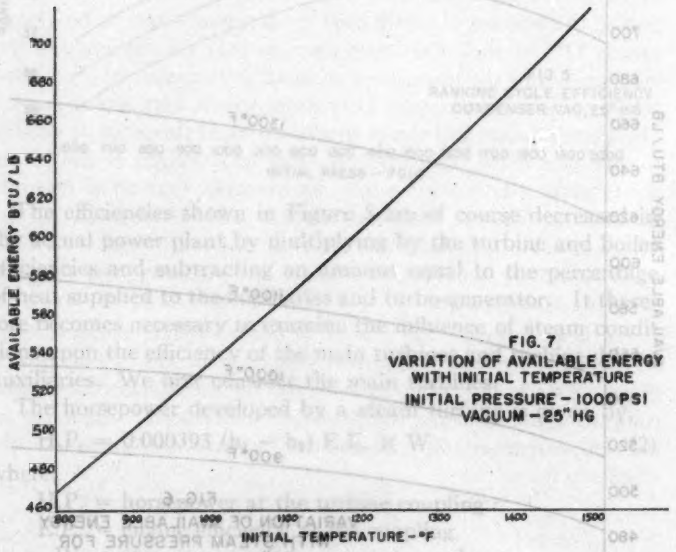


FIG. 7
VARIATION OF AVAILABLE ENERGY WITH INITIAL TEMPERATURE
INITIAL PRESSURE - 1000 PSI
VACUUM - 25" HG

power developed is directly proportional to the available energy, by formula (2), it is clear that more energy is available per pound of steam as a result of increasing initial temperature and pressure and by decreasing back pressure. However, initial and final steam conditions, as well as the turbine size and rotative speed, have an effect on the engine efficiency. This effect will be apparent by a consideration of turbine losses.

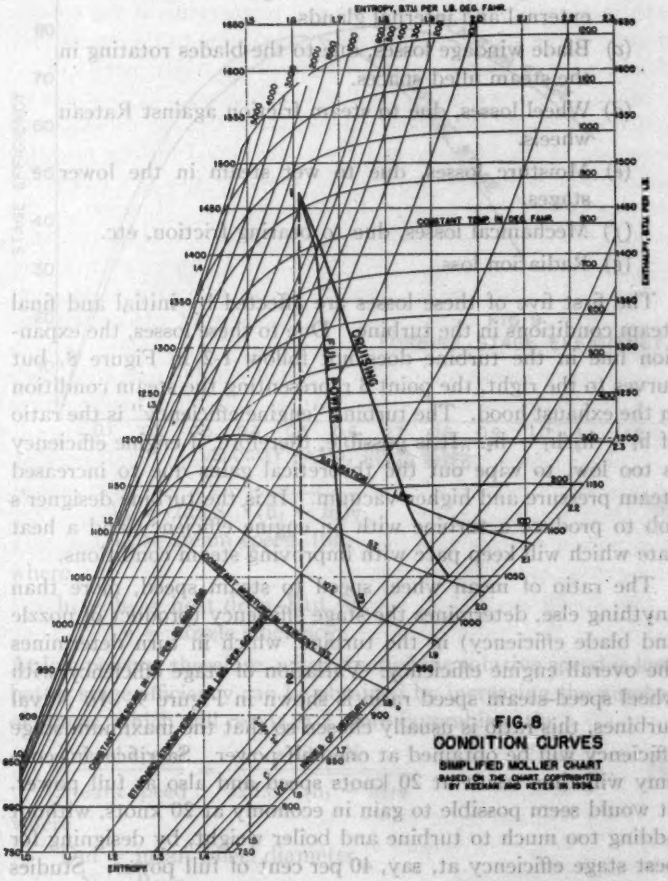


FIG. 8
CONDITION CURVES
SIMPLIFIED MOLLIER CHART
BASED ON THE CHART COPYRIGHTED BY HEDKAM AND KEYES IN 1924.

Losses in the turbine may be summarized as follows:

(*a*) Leaving loss, due to residual energy in the steam leaving the last row of blades.

(*b*) Leakage losses, due to leakage of steam at the external and internal glands.

(*c*) Blade windage losses, due to the blades rotating in the steam filled spaces.

(*d*) Wheel losses, due to steam friction against Rateau wheels.

(*e*) Moisture losses, due to wet steam in the lower stages.

(*f*) Mechanical losses, due to bearing friction, etc.

(*g*) Radiation loss.

The first five of these losses are affected by initial and final steam conditions in the turbine. Due to these losses, the expansion line in the turbine does not follow 1-2 in Figure 8, but curves to the right, the point 3 representing the steam condition in the exhaust hood. The turbine "engine efficiency" is the ratio of $h_1 - h_3/h_1 - h_2$. It is possible, therefore, if engine efficiency is too low, to wipe out the theoretical gains due to increased steam pressure and higher vacuum. It is the turbine designer's job to produce a turbine with an engine efficiency and a heat rate which will keep pace with improving steam conditions.

The ratio of mean wheel speed to steam speed, more than anything else, determines the stage efficiency (product of nozzle and blade efficiency) in the turbine, which in turn determines the overall engine efficiency. Variation of stage efficiency with wheel speed-steam speed ratio is shown in Figure 9. In Naval turbines, this ratio is usually chosen so that the maximum stage efficiency will be obtained at one-half power. Sacrifices in economy will thus result at 20 knots speed and also at full power. It would seem possible to gain in economy at 20 knots, without adding too much to turbine and boiler weight, by designing for best stage efficiency at, say, 40 per cent of full power. Studies along these lines would seem to be profitable.

Increasing the number of turbine stages decreases the heat drop per stage and thus the speed of the steam leaving the nozzles, since,

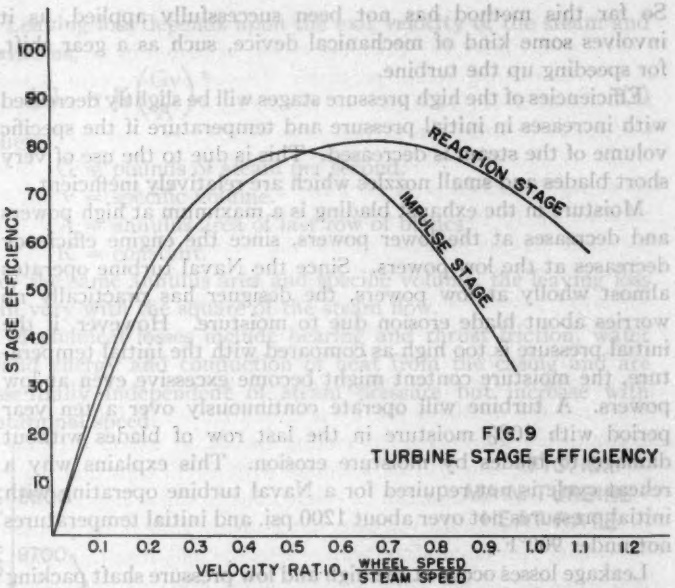


FIG. 9
TURBINE STAGE EFFICIENCY

$$V_s = 223.8\ \sqrt{(h_1 - h_2)\eta} \qquad (4)$$

V_s = steam speed, fps.

where

$h_1 - h_2$ = heat drop, Btu./lb.

η = nozzle efficiency.

At low powers, therefore, when the turbine rotative speed is low, better stage efficiency can be obtained by increasing the number of stages, which will result in a more favorable ratio,

$$\frac{\text{Wheel speed}}{\text{Steam speed}} = \frac{\pi\ \text{Dm N}}{223.8\ \sqrt{(h_1 - h_2)\eta}} \qquad (5)$$

where

Dm = mean wheel diameter.

N = Rpm.

The use of a cruising turbine, which amounts to adding stages to the high pressure end of the turbine at low speeds, has been common practice in the past. The same effect on turbine economy could be obtained by increasing wheel speeds at low powers.

So far this method has not been successfully applied, as it involves some kind of mechanical device, such as a gear shift, for speeding up the turbine.

Efficiencies of the high pressure stages will be slightly decreased with increases in initial pressure and temperature if the specific volume of the steam is decreased. This is due to the use of very short blades and small nozzles which are relatively inefficient.

Moisture in the exhaust blading is a maximum at high powers and decreases at the lower powers, since the engine efficiency decreases at the low powers. Since the Naval turbine operates almost wholly at low powers, the designer has practically no worries about blade erosion due to moisture. However, if the initial pressure is too high as compared with the initial temperature, the moisture content might become excessive even at low powers. A turbine will operate continuously over a ten year period with 10% moisture in the last row of blades without damage to blades by moisture erosion. This explains why a reheat cycle is not required for a Naval turbine operating with initial pressures not over about 1200 psi. and initial temperatures not under 900°F.

Leakage losses occur at the high and low pressure shaft packing glands, in the diaphragm labyrinths of impulse turbines and at the blade tips and dummy packing in reaction turbines. They vary almost directly with increase in pressure, but decrease with an increase in steam temperature.

Windage losses include disc friction and fanning action of idle blades, in the case of part-peripheral admission stages. They vary as follows:

$$\text{Loss} = \frac{KD^5N^3}{v} \qquad (6)$$

where

N = Rpm.
D = diameter of blade ring.
v = specific volume of steam.
K = constant.

The mean blade diameter has a greater effect than the revolutions and the specific volume. Considering the whole turbine, windage loss will not increase with increased initial temperature and pressure.

Leaving loss depends upon the exit velocity of the steam and varies as,

$$\text{L.L.} = K\left(\frac{Gv}{A}\right)^2 \tag{7}$$

where

G = pounds of steam per second.
v = specific volume.
A = annulus area of last row of blades.
K = constant.

For the same annulus area and specific volume, the leaving loss will vary with the square of the steam flow.

Mechanical losses include bearing and thrust friction, water gland energy and conduction of heat from the casing and are essentially independent of steam pressure but increase with rotational speed.

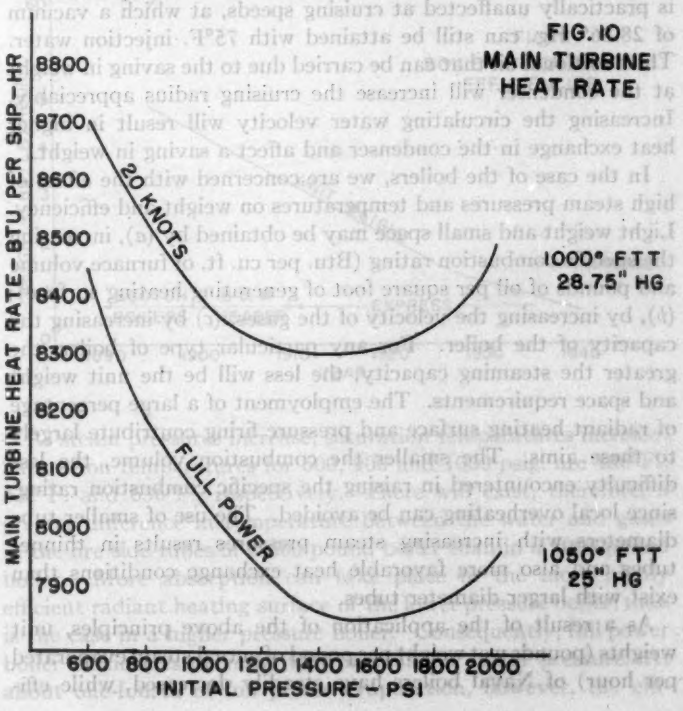


FIG. 10
MAIN TURBINE
HEAT RATE

The above complex set of relations may best be summed by referring to Figure 10, which shows the variation in turbine heat rate with varying steam pressure. The optimum point is seen to occur at about 1400 psi. for the 20 knot condition. Next to the velocity ratio, the volume flow (cubic feet per second) of steam is the chief factor influencing turbine efficiency. The efficiency increases with increase in volume flow for a given turbine, up to its designed capacity, due to a decrease in the per cent leakage losses. Since the Naval turbine is a high powered machine operating at low powers most of the time, moderate pressures and high temperatures, or large specific volumes, will result in improved engine efficiencies at low powers.

Calculations show that condenser surface is about halved in reducing vacuum from 27½″ Hg. to 25″ Hg. at full power. Although the available energy is thus reduced at full power, it is practically unaffected at cruising speeds, at which a vacuum of 28.86″ Hg. can still be attained with 75°F. injection water. The additional oil that can be carried due to the saving in weight at the condenser will increase the cruising radius appreciably. Increasing the circulating water velocity will result in higher heat exchange in the condenser and affect a saving in weight.

In the case of the boilers, we are concerned with the effect of high steam pressures and temperatures on weight and efficiency. Light weight and small space may be obtained by (*a*), increasing the specific combustion rating (Btu. per cu. ft. of furnace volume and pounds of oil per square foot of generating heating surface); (*b*), by increasing the velocity of the gases, (*c*) by increasing the capacity of the boiler. For any particular type of boiler, the greater the steaming capacity, the less will be the unit weight and space requirements. The employment of a large percentage of radiant heating surface and pressure firing contribute largely to these aims. The smaller the combustion volume, the less difficulty encountered in raising the specific combustion rating, since local overheating can be avoided. The use of smaller tube diameters with increasing steam pressures results in thinner tubes and also more favorable heat exchange conditions than exist with larger diameter tubes.

As a result of the application of the above principles, unit weights (pounds wet weight per pound of actual steam evaporated per hour) of Naval boilers have steadily decreased, while effi-

ciencies have increased for boilers operating in the 400-600 psi. pressure region. See Figure 11.

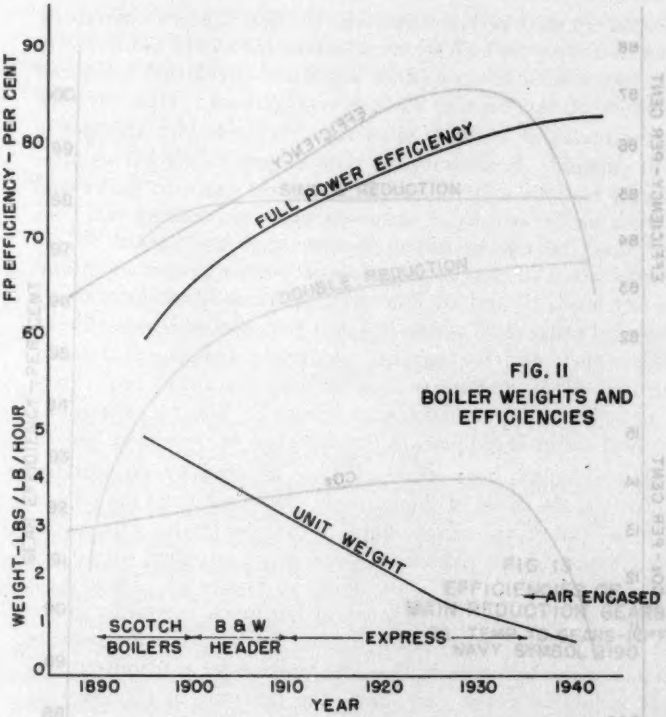


FIG. II
BOILER WEIGHTS AND EFFICIENCIES

As steam pressures increase, saturation temperatures increase. Saturation temperatures for 600, 900 and 1000 psig. are 486°F., 532°F. and 636°F., respectively. There will exist, therefore, a greater difference in temperature between the water and gases in the fire side tubes of a 600 pound boiler than in a 2000 pound boiler. More absorption can take place in the more highly efficient radiant heating surface of the lower pressure boiler, than is the case in a higher pressure boiler. Consequently, full power boiler efficiencies decrease with increase in steam pressure. At about one-fourth of full power evaporation, however, the effi-

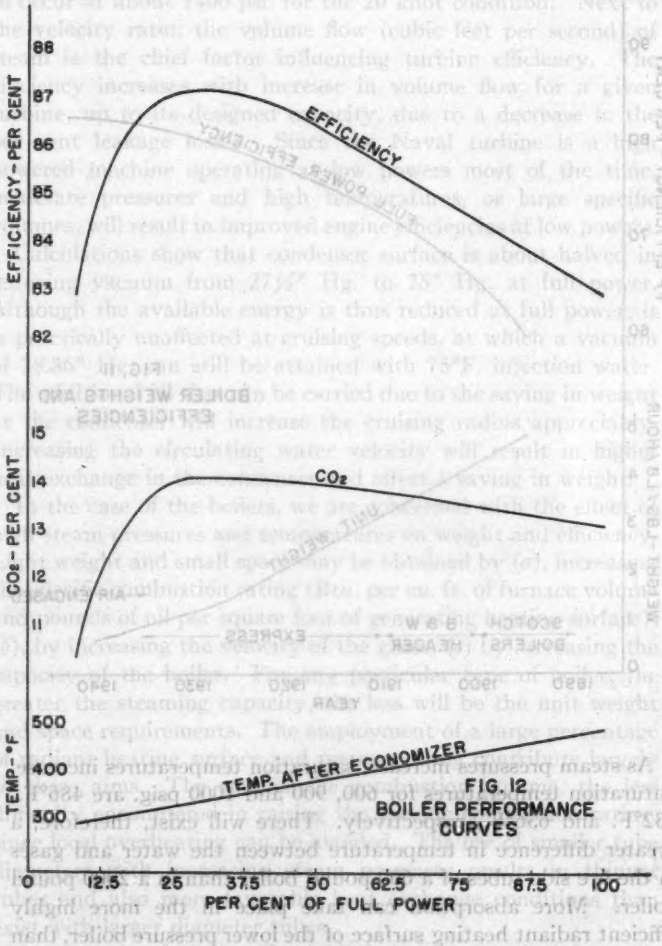


FIG. 12
BOILER PERFORMANCE CURVES

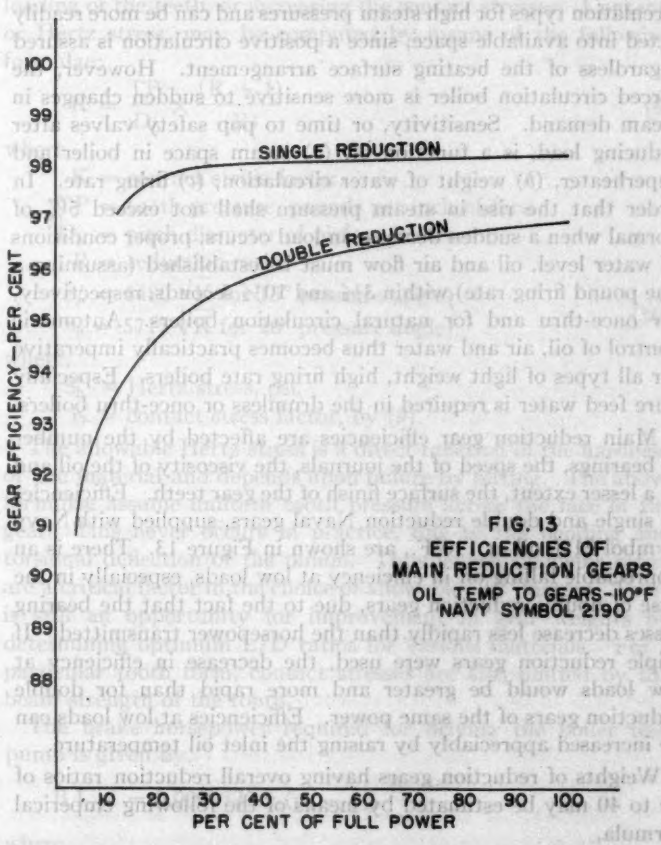


FIG. 13
EFFICIENCIES OF
MAIN REDUCTION GEARS
OIL TEMP TO GEARS-110°F
NAVY SYMBOL 2190

ciencies of modern Naval boilers are about the same regardless of operating pressure. Figure 12 shows the variation of efficiency with load for a Naval boiler designed for 600 #-850°F.T.T.

Forced circulation type boilers can be built lighter than natural circulation types for high steam pressures and can be more readily fitted into available space, since a positive circulation is assured regardless of the heating surface arrangement. However, the forced circulation boiler is more sensitive to sudden changes in steam demand. Sensitivity, or time to pop safety valves after reducing load, is a function of, (*a*) steam space in boiler and superheater, (*b*) weight of water circulation, (*c*) firing rate. In order that the rise in steam pressure shall not exceed 5% of normal when a sudden decrease in load occurs, proper conditions of water level, oil and air flow must be established (assuming a one pound firing rate) within 3½ and 10½ seconds, respectively, for once-thru and for natural circulation boilers. Automatic control of oil, air and water thus becomes practically imperative for all types of light weight, high firing rate boilers. Especially pure feed water is required in the drumless or once-thru boilers.

Main reduction gear efficiencies are affected by the number of bearings, the speed of the journals, the viscosity of the oil and to a lesser extent, the surface finish of the gear teeth. Efficiencies of single and double reduction Naval gears, supplied with Navy Symbol 2190 oil at 110°F., are shown in Figure 13. There is an appreciable falling off in efficiency at low loads, especially in the case of double reduction gears, due to the fact that the bearing losses decrease less rapidly than the horsepower transmitted. If triple reduction gears were used, the decrease in efficiency at low loads would be greater and more rapid than for double reduction gears of the same power. Efficiencies at low loads can be increased appreciably by raising the inlet oil temperature.

Weights of reduction gears having overall reduction ratios of 20 to 40 may be estimated by means of the following emperical formula,

$$W = 3.7\,[(\pi D_1 \times FW_1) + (\pi D_2 \times FW_2) + — \\ \pi D_n \times FW_n] \qquad (8)$$

where

$D_1\ D_2\ D_n$ = pitch diameters of gears and pinions, inches.

FW_1 FW_2 FW_n = face width of gears and pinions, inches.
W = weight in pounds.

The weight of a gear may be reduced by increasing the specific loading of the teeth, or increasing the contact stresses. Contact, or Hertz stress, may be computed by means of the following formulae:

$$K = \frac{TP}{D_p} \times \frac{(R + 1)}{(R)} \qquad (9)$$

where,

K = contact stress factor.
TP = tooth pressure, pounds per inch of face.
D_p = pitch diameter of pinion, inches.
R = reduction ratio.

$$\left.\begin{aligned} S_h &= 6480\sqrt{K} \text{ for } 15° \text{ pressure angle} \\ S_h &= 5715\sqrt{K} \text{ for } 20° \text{ pressure angle} \end{aligned}\right\} \qquad (10)$$

where,

S_h = Hertz stress, psi.
K = contact stress factor, by (9).

The allowable Hertz stress is a direct function of the hardness of the material and depends upon failure by pitting. The above formulae assume uniform tooth pressure across the face of the gear. This never occurs in practice, due to the bending and torsional deflection of the pinion. Thus length/diameter ratios are a critical factor in the choice of allowable "K" factors. There is still an opportunity for improvement in gear weights by determining optimum L/D ratios for various materials. For a particular tooth form, contact stresses are also limited by the beam strength of the tooth.

The brake horsepower required for driving the boiler feed pump is given by,

$$\text{B.L.p.} = 0.0000727\ v_1(p_2 - p_1)\frac{W}{e}$$

where,

p_1 = pump suction pressure, psi.
p_2 = pump discharge pressure, psi.
v_1 = specific volume of water at P_1, Cu. ft./lb.
W = pounds of water pumped per hour.
e = pump efficiency.

Feed pump work thus increases essentially in proportion to boiler pressure and varies between approximately 1% for 600 psi. and 3% for 2000 psi. boilers.

As gas velocities are increased in an effort to decrease the specific weights of boilers, the gas pressure drop is increased, which results in larger blowers. The brake horsepower required to drive a blower is given by,

$$\text{B.h.p.} = \frac{0.1572 \times A \times H}{e}$$

where,

H = head, inches of water.
A = thousands of cubic feet/min. of air.
e = blower efficiency.

Blower work is found to be about doubled for boilers of like capacity but delivering steam at 600 and 2000 psi., respectively.

The chief factor determining the upper limit of high temperature steam is the material available for piping, valves and high pressure turbine blading. The following properties are desired in a material suitable for high temperature steam service: (*a*) high rupture strength (stress rupture), (*b*) high creep strength, (*c*) good ductility, (*d*) high yield strength, (*e*) low coefficient of expansion, (*f*) good weldability. Since saving in weight is a prime consideration, piping for Naval service should be of a better grade material than is commonly used for stationary power plants. Carbon steel has accordingly been employed for temperatures up to 650°F., whereas shore stations use it for temperatures up to 800°F. Between 650°F. and 900°F., carbon-molybdenum is employed on Naval vessels. The addition of chromium in small amounts, about 1½ to 2¼% is desirable for piping in the 900°-1050°F. range. No experience is available for higher steam temperatures, although the oil refining industry has used austenitic steels for temperature approximating 1500°F.

It is desirable to keep pipe diameters as small as possible, not only because weights are decreased, but also because pipe stresses and end thrusts, due to thermal expansion, decrease with smaller diameters. Steam pressure drop from superheater outlet to throttle will be very small at cruising speeds, even though pipe diameters are chosen to give a pressure drop of ten per cent of boiler pressure at full power. However, if pipe sizes are too

small, the resulting excessive pressure drop at full power will necessitate an increase in the turbine, condenser and boiler sizes that will negate the weight saving in the piping. A decrease in available energy in the turbine at full power, due to increased pressure drop resulting from the use of smaller pipes, will result in an increase in the oil burned per unit of time. This must be balanced against the decrease in the pipe weight due to the use of the smaller size.

It has been common practice on Naval vessels to heat the feed water in only one feed heater by means of auxiliary exhaust. In case all of the auxiliaries are steam turbine driven, an excess of exhaust is available at the low powers, unless the evaporators are operating. This excess is "dumped" into the main condenser and the resultant loss of heat decreases the power plant efficiency. Reintroducing the excess exhaust steam into the low pressure turbine to do useful work has been tried on Naval vessels but introduces operating complications which are undesirable. It is obviously desirable to obtain a balanced cycle, i.e., one in which just enough auxiliary exhaust is available to heat the feed, at 20 knots. High pressure cycles operating at relatively low full power vacuum, will be balanced at 20 knots by employing fewer steam driven auxiliaries than are necessary if a higher full power vacuum is used. If only the main feed pump and blower turbines, for example, operate with high pressure steam, the exhaust from these will be superheated. In addition, the water rate of the main turbines is low if a high pressure-high temperature cycle is employed, so that a relatively small amount of condensate needs to be heated. Efficient operation of all other auxiliaries can be obtained by making these electric drive and installing at least one ship's service generator of such a capacity that it will operate at its most efficient load at 20 knots. Formerly, electric driven auxiliaries were not considered to be as reliable as steam driven, but this opinion no longer prevails.

Regenerative feed heating by bleeding from the main turbine and heating in several feed heaters, presents operating complications which prohibit its use on a Naval vessel. In addition, the extra weight of the feed heaters, piping and valves is objectionable. Furthermore, regenerative feed heating is of greatest advantage at full load operation and shows but slight gains in economy at low powers.

In order to show the effect of the efficiencies of boilers, main turbines, gears and auxiliaries on the fuel consumption of the power plant, comparison is made between two hypothetical cycles at full power operation. The first operates between 600 psi.-850°F.T.T. and 25″ Hg. vacuum and turbine, gear and boiler efficiencies are assumed to be 82%, 98% and 83%, respectively. The auxiliaries, including ship's service generator, use 3% of the total steam generated. The Rankine cycle efficiency will be 34.8% and the power plant efficiency will be $(34.8 \times .82 \times .98 \times .83 - 3) = 20.2\%$. The corresponding overall fuel rate is, assuming 18500 Btu. per pound of oil burned,

$$\text{Fuel rate} = \frac{2545}{18500 \times 0.203} = 0.68 \text{ pounds oil per Shp.-hr.}$$

Assume the second cycle to operate between initial steam conditions of 1200 psi.-1100°F. and a vacuum of 25″ Hg. at full power. Turbine, gear and boiler efficiencies are assumed to be 70%, 98% and 80%, respectively, and the auxiliaries again use 3% of the total steam generated. The Rankine cycle efficiency is now 39.6% and the actual power plant efficiency is, $(39.6 \times .70 \times .98 \times .80 - 3) = 18.7\%$. The power plant fuel rate is,

$$\text{Fuel rate} = \frac{2545}{18500 \times 0.187} = 0.737 \text{ pounds oil per Shp.-hr.}$$

From the above we draw the following conclusions:

(*a*) Increasing initial steam pressures and temperatures and increasing vacuum results is an improvement in Rankine cycle efficiency which may not be realized in the plant fuel rate if the efficiencies of the main turbines, gears, boilers and auxiliaries are adversely affected.

(*b*) An optimum steam pressure exists for the power plant of a Naval vessel which, if exceeded, will result in decreased radius of action at cruising speed.

(*c*) Increases in steam temperature offer almost proportionate improvement in main turbine economy.

(*d*) The type of drive used for the auxiliaries should be chosen so that a "balance" of the cycle exists at the most used cruising speed.

(*e*) An optimum full power vacuum exists which will result in a maximum radius of action on cruising speed, resulting from a greatly decreased condenser weight.

(*f*) Efficiency of the main turbines, gears and boilers is of greater importance than that of the auxiliaries.

(*g*) The chief factors contributing to poor turbine efficiency at low powers are an unfavorable wheel speed-steam speed ratio, and a reduction in volume flow of steam, which increases leakage losses. Improvement in reducing the latter has been made in the Roeder turbine, which utilizes a "breathing lung" inner casing that avoids blade and packing gland rubs during changing loads, so that very small clearances can be used.

(*h*) Boiler sensitivity increases as specific weight decreases. Unless reliable automatic control of air, oil and water is obtainable, light weight high pressure boilers will present more operating difficulties than the heavier drum type lower pressure boilers.

(*i*) Gear efficiencies may be increased at low powers by using excess auxiliary exhaust to heat the lubricating oil.

A METHOD OF DETERMINING THE NATURAL FREQUENCIES OF BENDING VIBRATION OF UNIFORM BEAMS.

By James A. Fay.

I. Introduction.

The determination of the natural frequencies of elastic structures and the critical speeds of rotating members often requires an elaborate analysis, inasmuch as such systems seldom can be represented by simple elastic systems for which unique solutions of the frequency equation are known. Whenever a structure can be considered to be composed of sections of uniform beams, a convenient method derived from the uniform beam vibration theory frequently permits the calculation of its natural frequencies with less effort.

Several methods are now widely used for determining the natural frequencies of structures. The Rayleigh method, based on the equivalence of the kinetic and potential energies of vibration, may be used to determine the fundamental frequency. It is only as accurate as the assumed deflection curve from which the energies were calculated, and always results in a frequency higher than the true frequency. A second method, originated by Stodola, is both exact and convergent, enabling the determination of all modes of vibration in succession beginning with the fundamental. It becomes quite tedious for the higher modes of vibration and for structures supported at several locations. (This method is commonly used in determining the natural frequencies of transverse vibration of a ship's hull.) A third method, devised by Myklestad[1], is a sequence calculation whereby the structure is represented as a grouping of concentrated weights on a massless beam. The method proposed herein is closely allied to the latter.

The sequence calculation, which has been used for some time in torsional vibration studies, consists in finding the deflections

1. See "A New Method of Calculating Natural Modes of Uncoupled Bending Vibrations of Airplane Wings and Other Types of Beams" by N. O. Myklestad, Journal of Aeronautical Sciences, April 1944, pp. 153-162.

and internal reactions of an elastic system for several assumed frequencies. If all external conditions are satisfied the assumed frequency is the natural frequency. It is a trial and error process, and the work involved depends upon the correctness of the assumed frequencies. The higher modes of vibration may be computed as easily as the fundamental. The sequence calculation possesses another advantage: it will indicate whether the natural frequency is higher or lower than any selected value. This is very convenient in determining the critical speeds of rotating machinery.

The proposed method, which applies to most structures and rotating members, is a sequence calculation derived from the vibration theory for uniform beams.[2] It enables the determination of all modes of vibration for any structure which may be considered as a combination of sections of uniform beams, concentrated weights, or elastic (but weightless) beams. The effects of elastic supports and gyroscopic moments are easily incorporated in the general procedure.

In determining natural frequencies and critical speeds, the proper method to use depends upon the accuracy desired and the work involved. Since most elastic systems must be simplified or idealized to allow of analysis, accuracy is affected by the degree to which the idealized system departs from the actual as well as by the method of analysis. While the brevity of the solution may be a determining factor, it is highly desirable to utilize the idealized system which most closely represents the actual if the mechanics of computation are not excessive. In this regard, the proposed method is a convenient adjunct to existing procedures.

II. Transverse Vibration of a Uniform Beam.

In the following analysis, the usual assumptions concerning elastic bodies are retained: the material is homogeneous and conforms to Hooke's law. In addition, the deflection of the beam due to shear is neglected and the existence of damping forces is disregarded.

Figure 1 illustrates the positive direction of bending moment, vertical shear, slope and deflection for a vibrating uniform beam.

2. A "uniform beam" is one whose weight and stiffness does not vary from section to section.

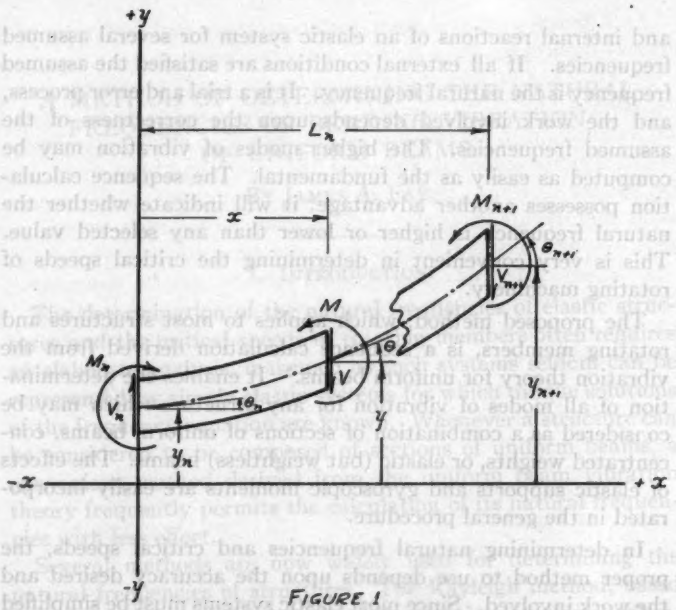


FIGURE 1

This convention of sign is important in the application to any problem.

The differential equation for the bending moment of the beam is:

$$M = EI \frac{\delta^2 y}{\delta x^2} \quad (1)$$

The vertical shear is

$$V = \frac{\delta M}{\delta x} = EI \frac{\delta^3 y}{\delta x^3} \quad (2)$$

Figure 2 shows the shear forces acting on the beam element "dx" of weight "wdx." Disregarding gravitational forces and forces external to the beam, its equation of motion is

$$\frac{w}{S} \frac{\delta^2 y}{\delta t^2} = - \frac{\delta V}{\delta x} = - EI \frac{\delta^4 y}{\delta x^4}$$

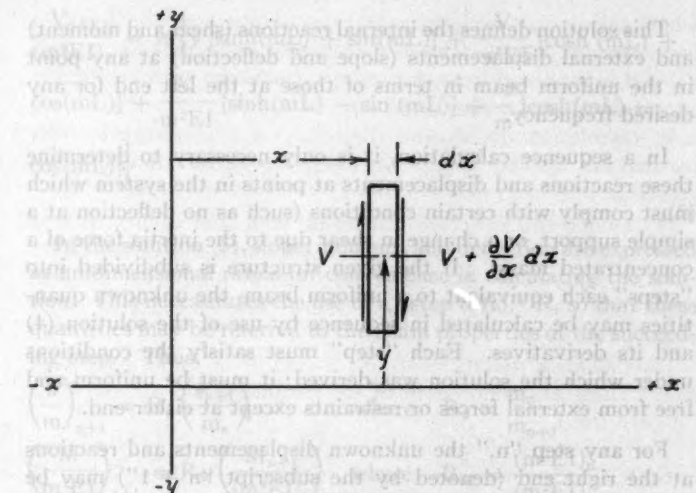


FIGURE 2

If the motion of the beam is harmonic of frequency "f", its deflection is expressed as

$$y = F(x) \cdot \cos(2\pi ft)$$

The equation of motion of the beam thus becomes

$$\frac{\delta^4 y}{\delta x^4} - m^4 y = o \qquad (3)$$

where

$$m^4 = \frac{4\pi^2 f^2 w}{SEI}$$

For a uniform beam, "m" is constant for any chosen frequency and a solution of (3) is

$$y = c_1 \sin mx + c_2 \cos mx + c_3 \sinh mx + c_4 \cosh mx \qquad (4)$$

The constants of integration are evaluated by substituting the end conditions at x = 0 in equations (1), (2), (4) and the slope equation

$$\theta = \frac{\delta y}{\delta x}$$

This solution defines the internal reactions (shear and moment) and external displacements (slope and deflection) at any point in the uniform beam in terms of those at the left end for any desired frequency.

In a sequence calculation, it is only necessary to determine these reactions and displacements at points in the system which must comply with certain conditions (such as no deflection at a simple support, or a change in shear due to the inertia force of a concentrated mass). If the given structure is subdivided into "steps" each equivalent to a uniform beam, the unknown quantities may be calculated in sequence by use of the solution (4) and its derivatives. Each "step" must satisfy the conditions under which the solution was derived: it must be uniform and free from external forces or restraints except at either end.

For any step "n," the unknown displacements and reactions at the right end (denoted by the subscript "n + 1") may be expressed in terms of those at the left end and the beam constants (denoted by "n") as follows (see Figure 1):

$$y_{n+1} = \tfrac{1}{2}\left\{\frac{\theta}{m}[\sinh(mL) + \sin(mL)] + y[\cosh(mL) + \cos(mL)] + \frac{V}{m^3EI}[\sinh(mL) - \sin(mL)] + \frac{M}{m^2EI}[\cosh(mL) - \cos(mL)]\right\}_n$$

$$\frac{\theta_{n+1}}{m_n} = \tfrac{1}{2}\left\{\frac{M}{m^2EI}[\sinh(mL) + \sin(mL)] + \frac{\theta}{m}[\cosh(mL) + \cos(mL)] + y[\sinh(mL) - \sin(mL)] + \frac{V}{m^3EI}[\cosh(mL) - \cos(mL)]\right\}_n$$

$$\frac{M_{n+1}}{(m^2EI)_n} = \tfrac{1}{2}\left\{\frac{V}{m^3EI}[\sinh(mL) + \sin(mL)] + \frac{M}{m^2EI}[\cosh(mL) + \cos(mL)] + \frac{\theta}{m}[\sinh(mL) - \sin(mL)] + y[\cosh(mL) - \cos(mL)]\right\}_n$$

$$\frac{V_{n+1}}{(m^3EI)_n} = \tfrac{1}{2}\left\{y\,[\sinh(mL) + \sin(mL)] + \frac{V}{m^3EI}\,[\cosh(mL) + \cos(mL)] + \frac{M}{m^2EI}\,[\sinh(mL) - \sin(mL)] + \frac{\theta}{m}\,[\cosh(mL) - \cos(mL)]\right\}_n \tag{5}$$

In the solution (5), slope, moment, and deflection are expressed as unidimensional ratios for convenience in calculating the solution. This necessitates the use of a "step ratio," R, so that these quantities may be referred to the beam properties of the succeeding step. Thus

$$\left(\frac{\theta}{m}\right)_{n+1} = R_\theta\left(\frac{\theta_{n+1}}{m_n}\right) \quad \text{where} \quad R_\theta = \frac{m_n}{m_{n+1}}$$

$$\left(\frac{M}{m^2EI}\right)_{n+1} = R_M\left(\frac{M_{n+1}}{[m^2EI]_n}\right) \quad \text{where} \quad R_M = \frac{(m^2EI)_n}{(m^2EI)_{n+1}}$$

$$\left(\frac{V}{m^3EI}\right)_{n+1} = R_V\left(\frac{V_{n+1}}{[m^3EI]_n}\right) \quad \text{where} \quad R_V = \frac{(m^3EI)_n}{(m^3EI)_{n+1}}$$

A more convenient representation of this solution is shown in Table I. In this tabulation provision is made for all quantities required in the calculations for each step, including the following:

(*a*) *Conditions of restraint*: Limitations imposed on deflection or slope at the right end.

(*b*) *External moment*: Inertia (gyroscopic) moment for rotating discs, or elastic foundation moment proportional to slope.

(*c*) *External shear force*: Inertia force of concentrated weight, or reaction of fixed or elastic support.

(*d*) *Step ratios*: Slope, moment, and shear at the right end are referred to the succeeding step when multiplied by the dimensionless step ratios R_θ, R_m, and R_v. These values may then be used directly in the calculations for the succeeding step.

III. The Sequence Calculation.

The application of the uniform beam vibration theory to structures of continuously distributed mass and elasticity is accomplished by the sequence calculation. By this method the displacements and reactions of the entire structure are calculated for several trial frequencies, the true frequency being that

which satisfies all external conditions of restraint and freedom from exciting forces. The displacements and reactions are calculated in sequence from step to step beginning with arbitrarily assumed values for the initial step.

In calculating the natural frequency of a structure by this method, it will be found convenient to adopt the following procedure:

(1) Divide the structure into steps such that each step is a uniform beam subject to external forces, moments, or restraints only at either end. For example, a simply supported uniform beam with a concentrated weight at any point on the span would require two steps.

(2) Calculate quantities ①, ⑧, ⑨ and ⑩ in Table I for each step. These are independent of the frequency and are the same for each trial.

(3) Select a trial frequency by any appropriate approximation.

(4) Calculate the remaining quantities ⑤ through ⑦ of Table I for each step using the assumed frequency. If the trigonometric functions ④ through ⑦ are plotted graphically, considerable work will be saved in selecting the proper values for each step.

(5) At the beginning of the first step, two of the four displacements and reactions (deflection, slope, moment, and shear) will be known explicitly or as functions of the remaining two. A third may be selected as unity, and the fourth represented by an algebraic symbol.

(6) Using these values, calculate the displacements and reactions at the right end.

(7) Apply the conditions of restraint, and add all external moments and shear forces.[1]

(8) Repeat (6) and (7) for each step.

(9) At the last step, calculate the additional external force (or moment) required to satisfy the conditions of restraint and all other end conditions. If the correct frequency was selected, this force (or moment) will be zero.

(10) If necessary, select a new frequency and repeat the calculations beginning with (4).

1. This may introduce a new algebraic symbol while defining explicitly the existing one. For example, the condition of no deflection at a simple support will define explicitly one symbol, while requiring a new one to represent the unknown support reaction. Only one algebraic symbol is required for any one step.

While Table I provides for all possible requirements at any step, only the more complicated systems involve all these contingencies. In most cases the work is considerably less, and the calculation for each step will average between fifteen or twenty minutes (either by slide rule or calculating machine). Since three trials are generally sufficient to establish the natural frequency, a close estimate of the time required by this method in comparison with another can be made beforehand.

IV. Critical Speed of Propeller and Shafting.

Exemplifying the usefulness of the sequence calculation in dealing with marine engineering problems is the critical frequency calculation for the propeller and shafting shown in Figure 3. The detailed calculations for the four steps at the natural frequency are given in Table II.

The equivalent ideal system used in the calculation is illustrated in Figure 4. The shafting is considered simply supported at the outboard and stern bearings, and cantilevered at the steady bearing. The propeller exerts not only a shear force proportional to the deflection but also a gyroscopic moment proportional to the slope. Using the convention of sign assumed for the deviation of the beam equation, the ratio of propeller inertia (shear) force to displacement and the ratio of gyroscopic moment to slope are both negative.

Table II is fairly representative of the amount of work involved in a sequence calculation and illustrates one shortcut which reduces the number of calculations. Since the condition of restraint in three of the four steps is zero deflection at the right end ($y_{n+1} = 0$), in each of these steps the first (or deflection) column was computed and summed. The condition of restraint then determined the unknown support reaction (S_n) at the left end and also the numerical value of the shear $(V_n/(m^3EI)_n)$. The remaining columns were then computed using this shear.

In some cases the outboard bearings are not rigid, but possess elasticity in one direction. Furthermore, the bearings exert some restraining moment against a change in slope. It is possible to include these effects in the sequence calculation if the elasticities are measured or computed, but the gain in accuracy is offset by other effects less susceptible to analysis (such as the mass of water entrained by the propeller).

STEP — n, $n+1$ $\quad f =$	① $\frac{m_n}{\gamma_f} = \sqrt[4]{\left(\frac{4\pi^2 w}{gEI}\right)_n}$	② $m_n =$	③ $(mL)_n =$	
④ $(\sinh mL + \sin mL)_n$	$\left(\frac{\theta}{m}\right)_n(\sinh mL + \sin mL)_n$	$\left(\frac{M}{m^2EI}\right)_n(\sinh mL + \sin mL)_n$	$\left(\frac{V}{m^3EI}\right)_n(\sinh mL + \sin mL)_n$	$(\gamma)_n(\sinh mL + \sin mL)_n$
⑤ $(\cosh mL + \cos mL)_n$	$(\gamma)_n(\cosh mL + \cos mL)_n$	$\left(\frac{\theta}{m}\right)_n(\cosh mL + \cos mL)_n$	$\left(\frac{M}{m^2EI}\right)_n(\cosh mL + \cos mL)_n$	$\left(\frac{V}{m^3EI}\right)_n(\cosh mL + \cos mL)_n$
⑥ $(\sinh mL - \sin mL)_n$	$\left(\frac{V}{m^3EI}\right)_n(\sinh mL - \sin mL)_n$	$(\gamma)_n(\sinh mL - \sin mL)_n$	$\left(\frac{\theta}{m}\right)_n(\sinh mL - \sin mL)_n$	$\left(\frac{M}{m^2EI}\right)_n(\sinh mL - \sin mL)_n$
⑦ $(\cosh mL - \cos mL)_n$	$\left(\frac{M}{m^2EI}\right)_n(\cosh mL - \cos mL)_n$	$\left(\frac{V}{m^3EI}\right)_n(\cosh mL - \cos mL)_n$	$(\gamma)_n(\cosh mL - \cos mL)_n$	$\left(\frac{\theta}{m}\right)_n(\cosh mL - \cos mL)_n$
⑧ $R_\theta = \frac{(m)_n}{(m)_{n+1}}$	SUM	SUM	SUM	SUM
⑨ $R_M = \frac{(m^2EI)_n}{(m^2EI)_{n+1}}$	$Y_{n+1} = \frac{1}{2} \times SUM$	$\left(\frac{\theta}{m}\right)_{n+1} = \frac{1}{2} \times R_\theta \times SUM$	$\left(\frac{M}{m^2EI}\right)_{n+1} = \frac{1}{2} \times R_M \times SUM$	$\left(\frac{V}{m^3EI}\right)_{n+1} = \frac{1}{2} \times R_V \times SUM$
⑩ $R_V = \frac{(m^3EI)_n}{(m^3EI)_{n+1}}$	CONDITIONS OF RESTRAINT :		$\frac{\text{EXTERNAL MOMENT}}{(m^2EI)_{n+1}}$	$\frac{\text{EXTERNAL FORCE}}{(m^3EI)_{n+1}}$
			TOTAL	TOTAL

TABLE I

STEP – 0.1	$f = 6.75$	$\frac{m_0}{Ff} = 0.00271$	$m_0 = .00704$	$(mL)_0 = 1.267$ (72.6°)
2.5896	0	-1.0906	2.5896	0
2.2164	0	0	-0.9334	2.2164
0.6816	0.6816	0	0	-0.2871
1.6184	$1.6184\ M_0$	1.6814	0	0
$R_\theta = 1$	$0.6816 + 1.6184\ M_0$	0.5278	1.6562	1.9293
$R_M = 1$	$Y_1 = 0$	$\left(\frac{\theta}{m}\right)_1 = 0.2639$	$\left(\frac{M}{m^2EI}\right)_1 = 0.8281$	$\left(\frac{V}{m^3EI}\right)_1 = 0.9647$
$R_V = 1$	CONDITIONS OF RESTRAINT. $M_0 = \frac{-0.6816}{1.6184} = -0.4212$		0	S_1
				$0.9647 + S_1$

TABLE II

STEP – 1,2	$f = 6.75$	$\frac{m_1}{W_1} = 0.00271$	$m_1 = .00704$	$(mL)_1 = 3.379$ (193.6°)
14.423	3.8067	11.946	-16.246	0
13.721	0	3.6208	11.362	-15.456
14.894	$14.367 + 14.894\,S_1$	0	3.9302	12.333
15.663	12.970	-17.643	0	4.1333
$R_0 = 1$	$31.144 + 14894\,S_1$	-2.0762	-0.9538	1.0103
$R_M = 1$	$\gamma_2 = 0$	$\left(\frac{\theta}{ml}\right)_2 = -1.0381$	$\left(\frac{M}{m^2EI}\right)_2 = -0.4769$	$\left(\frac{V}{m^3EI}\right)_2 = 0.5052$
$R_V = 1$	CONDITIONS OF RESTRAINT: $S_1 = \frac{-31.144}{14.894} = -2.0911$		0	S_2
	$0.9647 + S_1 = -1.1264$			$0.5052 + S_2$

TABLE II

STEP – 2, 3	$f = 6.75$	$\frac{m_2}{f} = 0.00271$	$m_2 = .00704$	$(mL)_2 = 2.957$ (169.4°)
9.757	-10.133	-4.6526	15.783	0
8.643	0	-8.9757	-4.1214	13.981
9.391	$4.745 + 9.391\,S_2$	0	-9.7522	-4.4779
10.609	-5.0589	17.161	0	-11.017
$R_e = 1$	$-10.447 + 9.391\,S_2$	3.5327	1.9094	-1.5139
$R_M = 1$	$y_3 = 0$	$\left(\frac{\theta}{m^2}\right)_3 = 1.7664$	$\left(\frac{M}{m^3EI}\right)_3 = 0.9547$	$\left(\frac{V}{m^3EI}\right)_3 = -0.7569$
$R_V = 1$	CONDITIONS OF RESTRAINT:		0	S_3
	$S_2 = \frac{10.447}{9.391} = 1.1124$			$-0.7569 + S_3$
	$0.6052 + S_3 = 1.6176$			

TABLE II

STEP – 3,4	$f = 6.75$	$\frac{m_3}{W_3} = 0.00271$	$m_3 = .00704$	$(mL)_3 = 0.4224$ (24.2°)
0.8445	1.4317	0.8062	$-0.6395 + 0.8445 S_3$	0
2.0025	0	3.5371	1.3116	$-1.5163 + 2.0025 S_3$
0.0245	$-0.0186 + 0.0245 S_3$	0	0.0432	0.0234
0.1785	0.1704	$-0.1352 + 0.1785 S_3$	0	0.3153
$R_0 =$	$1.6435 + 0.0245 S_3$	$4.2081 + 0.1785 S_3$	$1.3153 + 0.8445 S_3$	$-1.1776 + 2.0025 S_3$
$R_M =$	$\frac{\gamma_4}{W_4} = 0.8218 + 0.0123 S_3$	$\frac{\theta_4}{m_3} = 2.1041 + 0.0893 S_3$	$\frac{M_4}{(m^2EI)_3} = 0.6577 + 0.4223 S_3$	$\frac{V_4}{(m^3EI)_3} = -.5880 + 1.0013 S_3$
$R_V =$	CONDITIONS OF RESTRAINT: $\frac{M_4}{(m^2EI)_3} = \frac{-286{,}000 f^2}{(mEI)_3}\left(\frac{\theta_4}{m_3}\right) = -0.0684 - 0.0029 S_3$		$\frac{-M_4}{(m^2EI)_3} = 0.0684 + 0.0029 S_3$	$\frac{-V_4}{(m^3EI)_3} = 2.3650 + 0.0355 S_3$
	$\frac{V_4}{(m^3EI)_3} = \frac{1228 f^2}{(m^3EI)_3}\left(\frac{\gamma_4}{W_4}\right) = -2.3650 + 0.0355 S_3$		$0 = 0.7261 + 0.4252 S_3$	$0 = 1.7762 + 1.0368 S_3$
			$S_3 = \frac{-0.7261}{0.4252} = -1.71$	$S_3 = \frac{-1.7762}{1.0368} = -1.71$

TABLE II

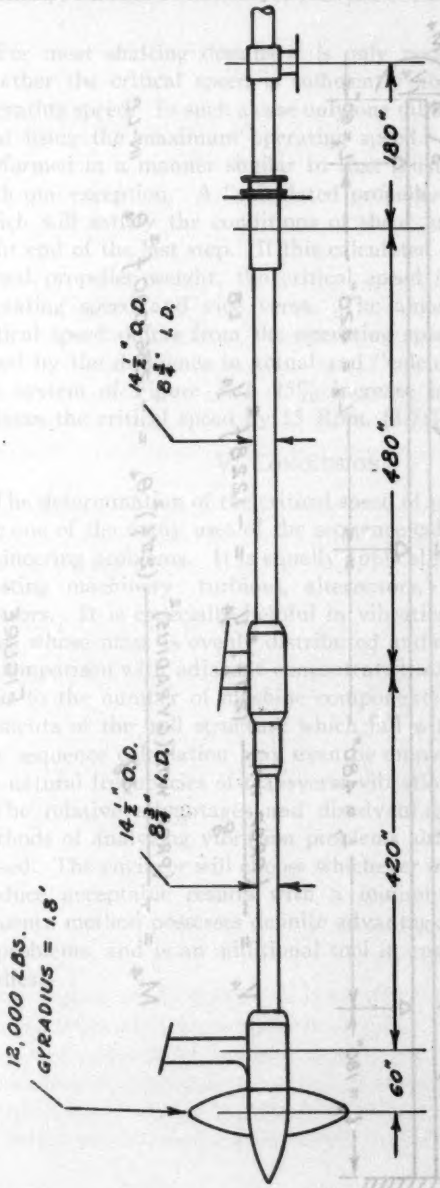
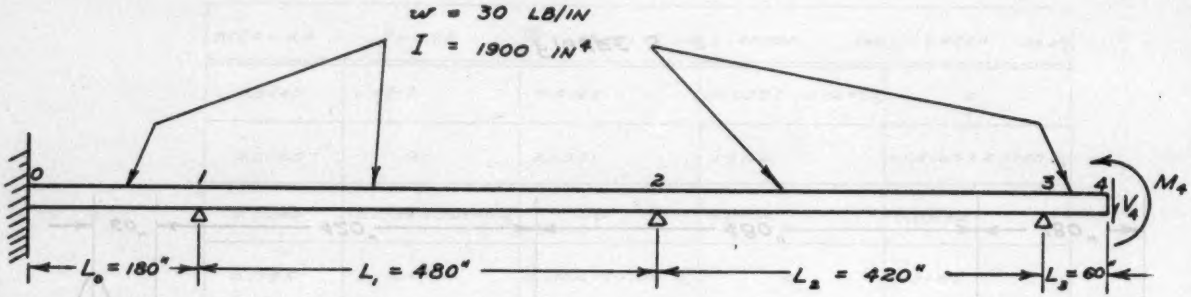

180"
480"
420"
60"
14 1/2" O.D.
8 3/4" I.D.
14 1/2" O.D.
8 3/4" I.D.
12,000 LBS.
GYRADIUS = 1.8'

FIGURE 3



$$V_4 = -\frac{(\text{PROP. WT.})}{g}(2\pi f)^2 Y_4 = -1228 f^2 Y_4 \quad \text{LB}$$

$$M_4 = -\frac{(\text{PROP. WT.})(\text{GYRADIUS})^2}{2g}(2\pi f)^2 \theta_4 = -286{,}000 f^2 \theta_4 \quad \text{IN-LB}$$

FIGURE 4.

For most shafting designs it is only necessary to ascertain whether the critical speed is sufficiently above the maximum operating speed. In such a case only one tabulation is necessary: that using the maximum operating speed. The calculation is performed in a manner similar to that illustrated in Table II, with one exception. A "calculated propeller weight" is chosen which will satisfy the conditions of shear and moment at the right end of the last step. If this calculated weight exceeds the actual propeller weight, the critical speed is higher than the operating speed and vice versa. The amount by which the critical speed differs from the operating speed is roughly indicated by the difference in actual and "calculated" weight. In the system of Figure 3, a 25% increase in propeller weight reduces the critical speed by 15 Rpm. (3.7%).

V. Conclusion.

The determination of the critical speed of propeller shafting is but one of the many uses of the sequence calculation in marine engineering problems. It is equally applicable to many types of rotating machinery: turbines, alternators, pumps, and compressors. It is especially helpful in vibration studies of structures whose mass is evenly distributed and not inconsequential in comparison with adjacent concentrated masses. There is no limit to the number of machine components, foundations, and segments of the hull structure which fall within this category. The sequence calculation may even be employed in computing the natural frequencies of transverse vibration of a vessel.

The relative advantages and disadvantages of the various methods of analyzing vibration problems already has been discussed. The engineer will choose whichever method promises to produce acceptable results with a minimum of labor. The sequence method possesses definite advantages for certain types of problems, and is an additional tool in engineering vibration studies.

IMPORTANCE OF DESIGN FACTORS FOR MARINE REDUCTION GEARS.

By W. G. Stoeckicht.

During the past 30 years considerable attention has been paid to research work concerning stresses in gears. Beam strength, contact stresses and the conditions causing scuffing have been carefully studied. Further investigations are being made with regard to the relation between stresses and endurance. But there is still contradiction between some theories and as far as their application is concerned, much has been left to the judgment of the individual designer.

A few years after World War I gear designers and manufacturers were surprised when they learned that gears had to be calculated not only for beam strength, but also for contact pressure.

The stresses deriving from contact pressure between two spheres or two rollers can be calculated by means of Hertz' equations. Two meshing teeth can be considered to be two rollers at the contact line. The radii of the rollers are then equal to the radii of curvature of the teeth at the point of contact. At the pitch line, the radius of curvature of each of the mating teeth is given by the formula,

$$R_c = R_p \sin \alpha,$$

in which:

R_c = radius of curvature.
R_p = radius of pitch circle.
α = pressure angle.

This formula clearly shows the influence of the pressure angle upon the size of the radii of curvature of a given gear.

Hertz' equations found their first and most important application in the design of ball and roller bearings. The allowable contact stress determines the maximum specific pressure in the zone of contact. In 1907 Stribeck introduced a new factor for

measuring contact stresses, which he called K-factor. For contact of two rollers this K-factor is defined by the formula

$$K = \frac{P}{l \cdot d_r}$$

in which

P = load in kgs.

d_r = relative diameter of curvature in cm., defined below.

l = length of line of contact in cm.

The relative diameter of curvature is defined by the formula

$$\frac{l}{d_r} = \frac{l}{d_1} + \frac{l}{d_2}$$

in which

d_r = relative diameter of curvature.

d_1 and d_2 = diameter of curvature of the two rollers, respectively.

The relation between the maximum contact stress in the zone of contact according to the Hertz formula and Stribeck's K-factor for steel is given by the formula

$$p_{max} = 877^2 \sqrt{K}$$

in which

p_{max} = maximum contact stress in kg./cm.^{2}

or

$$p_{max} = 3300^2 \sqrt{K}$$

in which loads are measured in lbs. and lengths in inches.

Stribeck's K-factor became very popular and was largely used in the ball and roller bearing industry and later in the gear industry. Apparently it gives more practical figures for comparison than the figure of p_{max} according to the Hertz formula. In the U. S. the K-factor has been introduced for gears in a modified form which neglects the influence of the pressure angle on the diameter of curvature and which therefore can be used only for the comparison of gears working under the same pressure angle.

Contact stresses were considered in regard to surface endurance. In the beginning it was believed and commonly accepted that for every material there was a specific contact stress, which, if not exceeded would guarantee practically indefinite life. Such assumption was backed by the results which Woehler obtained

when he carried out his laboratory tests concerning tensile and bending fatigue limits of various kinds of steel and other materials. For steel he found that a steel which stands 10^7 cycles of a given stress would stand it practically indefinitely. However, the assumption did not seem to be correct, either for contact or for bending stresses, as far as gear teeth are concerned. Recently, the author had the opportunity of examining gear wheels of German electric locomotives which had been in service for a period of 20 years. Some of these wheels showed cracks on several teeth at the tooth root of the sort which is typical for failures due to bending stresses. The fact that these gears performed their duty for a period of 20 years must be considered as being an excellent result; on the other hand, the failure proved also, that for longer periods of service there must be a relation between stress and endurance, which sets a limit to the latter for given stresses.

In the ball and roller bearing industry research work investigating the relation between contact stress and fatigue limit has been carried out on a very large scale. The Swedish SKF-concern probably has to be mainly credited for the very valuable results obtained. These seem to show a clear cubic relation between endurance and K-factor. Here we find an explanation why the use of the K-factor is more popular than that of p_{max}. If the relation between endurance and K-factor is of the third dimension, then endurance and p_{max} are related in the sixth dimension, and apparently it is easier to figure out a cubic relation than a sixth dimension relation when comparing two factors.

Two reasons contributed to the fact that the research work done by the ball and roller bearing industry produced reliable results. One of them was the very large number of tests which they were able to carry out on absolutely identical units. This large number of tests enabled them to also investigate statistically the amount of scattering of the values obtained. The other reason was that all tests were made using the same material in the same condition of hardness and tensile strength.

The question arises whether the results obtained on roller bearings can be applied to gears. In principle, there ought to be no reason why the material should react differently under contact stresses in gears and in roller bearings. Still two points have to be considered. When two gear teeth mesh, rolling motion

is combined with sliding motion, excepting on the pitch line. The sliding motion increases with the increasing distance of the point of contact from the pitch line. Sliding motion under pressure creates an additional tangential force at the point or line of contact. In a great number of tests which the author has carried out on friction gears, where at the point or line of contact both normal and tangential forces are working, he found that the additional tangential forces considerably increased the contact stresses and lowered life expectancy. But in lubricated gears pitting as a result of contact stresses always begins on the pitch line where there is no sliding motion. From this it may be concluded that the influence of tangential forces due to sliding motion can be neglected, and that therefore the results obtained from roller bearing tests may also be applied to gears.

The other point of difference may be seen in the fact that not all gears are provided with hardened teeth. When two spheres or two rollers contact under load, very high figures for compressive stresses are obtained according to Hertz' formula. In the case of high specific loads compressive Hertz stresses appear to be safe while the same material would not stand these stresses if the loads were to be applied so as to produce a plain state of stress. This phenomenon has been explained by the well-known fact that a crystal submerged in a fluid can be exposed to practically infinite compressive stress without altering its structure. Now this is the situation for the parts of the material exposed to the maximum Hertz stresses. It explains why steel of martensite structure shows the best results for taking up Hertz stresses. Indeed, the load carrying capacity (from the point of view of contact stresses) rapidly decreases with the decrease in hardness of steel. It is believed that the allowable K-factor varies as the square of the Brinell or Rockwell hardness of the steel used. So we have to deal with an uncertain factor in applying the results of roller bearing research to gears with non-hardened teeth. The picture will be further distorted by the fact, that with some non-hardened steels, as used in gears with soft teeth, a certain amount of work-hardening is encountered, which will be very difficult to account for.

Whilst it may be said therefore that an established law exists between contact stresses and endurance for hardened gears, this

relationship for soft gears is little more than an intelligent estimate.

It is a commonly accepted fact that hardened steel of martensite structure is the best obtainable material for surfaces working under Hertz stresses, and no one would even think of using or producing roller bearings made of soft steel. So it seems to be logical that the use of hardened load carrying gears is steadily increasing and it is to be expected that after a certain period of further evolution soft load carrying gears will be as rare as non-hardened ball or roller bearings are now. The available processes for steel hardening do not involve an obstacle for such a development, nor do we miss the facilities for grinding hardened gears. For the latter process we are indebted to the large amount of successful pioneer work done by Messrs. Maag in Zurich, Switzerland.

Dealing with the beam strength of gear teeth, we find that the importance of properly designed fillet radii was realized early. Quite as important however as the shape of the root fillet is its surface finish, particularly when hardened gears are used. Although the increase of the fatigue limit due to proper design and finish of tooth root fillets must not be overrated—some very careful comparative tests have shown an increase of allowable load of only 15 to 20%—these improvements probably eliminate many hazards and allow for a more dependable calculation of endurance with regard to bending stresses.

Since the designer is able to calculate more or less exactly the probable endurance of a gear from the point of view of contact stresses and of beam strength, he is then faced with the problem of how to balance both stresses in his design (the influence of the factor controlling scuffing can be neglected in this connection, because preventing of scuffing is a yes or no question with no immediate relation to endurance). It is quite natural that one should feel inclined to balance both stresses as far as possible. This, in general, will lead to the use of a comparatively fine pitch which from some points of view will be claimed as an advantage. This theory of balance of stresses, with approximately the same number of hours of life both for contact and bending endurance, seems to be widely accepted. And gear designs which do not agree with this rule are readily declared as poor.

The author believes that this rule is entirely wrong. It may give to the calculation of a gear the satisfying aspect of an esthetical harmony, but this is all that can be claimed in its favor. The adherents to that rule do not consider two important facts:

1. That the equations by which the endurance can be calculated are statistical laws, this means that the results obtained by the calculation do not represent more than the probability of a certain percentage of accuracy. This scattering in the results is particularly great in the zone of high stresses and, therefore, low figures for hours of life.

2. That the endurance figures for beam strength and contact stresses have an entirely different importance for the safety of a gear and the machinery connected with it.

Let us assume that a gear fails because its endurance limit due to contact stresses has been reached. This means that pitting begins to show, but it does not mean a sudden breakdown. The gear may go on running for a considerable time. If, for instance, this occurs on board ship, nothing will happen and in most cases the ship will be able to continue its voyage until the set of gears can be exchanged, particularly when the further performance will be at partial loads.

Things are different when the endurance limit is reached from the point of view of beam strength. Then tooth breakage occurs which is always a breakdown of the whole unit. If for instance it happens aboard a single screw ship, this vessel will float helplessly on the ocean.

An investigation of the failure will disclose that the gear had been properly designed according to the rules. The designer will be backed by the faculty which believes in the rules, and it can be expected that the material will be blamed, or errors in heat treatment will be accepted.

Also the effects of occasional overloads are more detrimental from the point of view of beam strength than from that of contact stresses. Such overloads will effect contact stress endurance only in the way that they will abbreviate the life of the gear, but this will be noticeable only when the overload will be applied for a comparatively long period. On the other hand, overloads in gears with normally high bending stresses can easily lead to

the initiating of root fillet cracks which later will result in breakage.

For these reasons the basic rule for gear design ought to be that a gear never should fail through tooth breakage. The calculated life expectancy from the point of view of bending stresses ought to be so far greater than the endurance with regard to contact stresses that no gear will have to be replaced for complete breakdown. Such a rule may sound heretical, but it will have the best chances of satisfying the requirements of the customer.

In 1931 Hofer published a formula concerning the relation between the surface of a pinion and the number of horsepower it can transmit continuously:

$$SA = \frac{m \times z \times b}{20N}$$

in which

N = horsepower transmitted.
m = module in m.m.
b = face width in m.m.
z = number of teeth.

Measuring lengths in inches, the formula would have to be written as follows:

$$SA = 3.23 \frac{D \times FW}{N}$$

in which

N = horsepower transmitted.
D = pitch line diameter in inches.
FW = face width in inches.

Hofer said that if the value of the SA-factor becomes less than 1 the pinion will be subject to premature wear. This wear is supposed to be due to heat accumulation. He applied his formula to hardened gears; non-hardened gears indeed will be nearly always on the safe side because, due to their comparatively low allowable stresses, they have rather large dimensions.

Even considering hardened gears, the problem of applying Hofer's formula as a rule will be encountered in gears operating at high speeds. There has been much discussion about it, and it has been ascertained that safe performance has very often been achieved with SA-factors very much smaller than 1, but

there was a great divergence of opinion as to how far one can go. For some applications, for instance, an SA-factor of 0.4 was considered to be safe.

The formula ought not be used in connection with scuffing. The conditions which cause this kind of wear can be analyzed through special formula. But undoubtedly the SA-factor is an important aid for the consideration of life expectancy with regard to contact stresses. It may be remembered that the endurance figures for ball and roller bearings have to be reduced when working temperatures exceed certain limits. Through many years the author has paid much attention to the extent to which allowable K-factors can be influenced by the SA-factor. In some cases he has found clear evidence for the fact that a very small SA-factor was responsible for pitting. He had found before that similar experience could be made with friction gears using friction members made of hardened steel. An attempt was made to improve working conditions by using austenitic steels which kept hardness up to comparatively high temperatures; but unfortunately these steels had a low heat conductivity, so that a higher accumulation of heat was obtained which prevented the desired effect. Similar experiences seem to have occurred in some part of the ball bearing industry. With the increasing tendency toward compact designs and high load carrying capacity, particularly in connection with high speeds, Hofer's SA-factor may become important for gear design, and it is to be hoped that valuable research work may be done with regard to its influence on durability.

NOTES.

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—Institute of Marine Engineers Transactions, April 1948.

THE EFFECT OF NON-CONTACT EXPLOSIONS ON WARSHIP MACHINERY DESIGN.
—Institution of Naval Architects, March 18, 1948.

LONG RANGE INDUSTRIAL PLANNING.
—Mechanical Engineering, January 1948.

RADIOGRAPHY AS A CONTROL FOR WELDING JOINTS IN PIPE LINES.
—The Welding Journal, June 1947.

LONGITUDINAL VIBRATION OF MARINE PROPELLER SHAFTING.

The following paper by Mr. C. P. Rigby appeared in the Institute of Marine Engineers Transactions for April 1948. It gives interesting details of efforts in Great Britain to reduce the vibration difficulties which arise from propeller excited vibrations in highly powered vessels. It provides further information on a subject of great importance already discussed in more general terms in the paper by Dr. F. H. Todd which was reprinted in the Notes of the JOURNAL of February 1948.

SYNOPSIS.

Longitudinal vibration of the propeller shafting has been the cause of seizure, serious scoring and excessive wear in the flexible couplings of Naval vessels and has in some cases given rise to unacceptable vibration of the bridge structure. Other defects such as difficulty in retaining stern gland packing, wear and loosening of stern tube bushes and loosening of rivets in thrust block seats have been attributed to the same cause. In three- and four-shafted ships the vibration is aggravated when turning because the blades of the inner propeller on the outside of the turn cut into the slip stream of the outermost wing propeller. It has been necessary on this account, in the majority of big ships, to reduce the power on the outermost wing shaft when turning at high speed, a procedure which introduces difficulties in control and reduces manoeuvrability.

The present paper describes how theory suggested that the vibration was magnified by resonance between the natural frequency of the system and the impulses arising from the number of blades on the propeller, and predicted that the amplitude would be reduced to an acceptable figure by changing the number of blades. Details are given of the vibration trials carried out to confirm the theory, and it is seen that the predicted improvement is, in fact, achieved. The trial results are worked up to give fundamental data as to thrust block flexibility, entrained water, thrust variation, damping factor, and the effect of turning. Using this information complete calculations are set out for a particular line of shafting, showing the effect of two possible thrust block positions and of using three- or five-bladed propellers. A second example treats the case of a very long shaft and shows that there is a need for further data. Using the worked examples as a basis for discussion, general principles are suggested by the application of which trouble may be cured in existing ships and avoided in new construction.

INTRODUCTION.

During the last five years a considerable amount of work, both theoretical and experimental, has been devoted to this problem. Experimental work has included the measurement of vibration during sea trials of ten ships and a static deflection test of a thrust block on shore under full thrust load. Theoretical work has consisted of analyzing trial results to obtain fundamental data and then using that data to predict the behavior of the shafting in new designs or the effect of modifications proposed for existing ships. Marine engineers will appreciate that when a ship, and more expecially a warship, is in service, there is a considerable lapse of time between the expression of a theory and its translation into fact; for example, over two years passed between the first proposal to fit a five bladed propeller and its sea trials. Partly for this reason there are still gaps to be filled in and this paper must, therefore, be regarded in some aspects as an interim report. The author hopes, however, that even with these shortcomings, it will prove both interesting and useful.

EARLY HISTORY.

In 1937 H.M.S. *Warspite* completed an extensive reconstruction which had included replacement of the original direct drive machinery by modern geared turbines. The new engines proved satisfactory in all normal respects until

steering trials commenced, when the claw type flexible couplings between the turbines and pinions of the inner shafts suffered serious damage. The damage consisted of scoring, rapid wear and even seizure of the mating claw faces. The first action was to increase supply of oil to the couplings, but further trials showed little improvement and it was concluded that excessive loads were being applied by vibration. After a most comprehensive investigation it was eventually discovered that a heavy longitudinal vibration occurred in the inner shaft on the outside of a turn and that this vibration involved rapid fore and aft sliding of the coupling claw faces. It was concluded that the cause of the vibration lay in the fact that with the stern of the ship swinging round the inner propeller was working partly in the wake of the outer screw, the latter being located 24 feet further forward. This condition is illustrated in Figure 2.

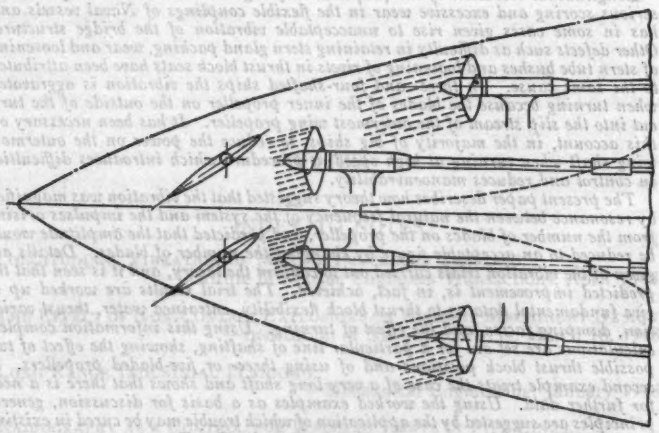

FIG. 2.—*Propeller interaction*

Further trials were then carried out with the object of discovering how to avoid the condition arising. It was eventually concluded that the only effective method was to eliminate the slip stream of the wing propeller on the outside of the turn by almost closing the main throttle valve of that set of machinery when turning at high speeds. This procedure was found to reduce the amplitude of vibration by one half, from an average maximum of ±.050 in. to ±.025 in.

In addition to the flexible coupling damage, there was heavy hull vibration in the aft portion of the ship, and serious leaks developed in the region of the propellers. The "easing" procedure described above prevented the recurrence of both troubles.

Following this experience *Queen Elizabeth* and *Valiant*, then being re-engined, were fitted with oversize main thrust blocks, and at the same time the "easing" rules were applied to all battleships except the twin screw *Rodney* and *Nelson*.

LATER HISTORY.

The trials of *Warspite* were not followed up by any further investigation, presumably because the easing procedure was so successful, but in new construction some consideration was given to the fore and aft distance between

FIGURE 1.—WORN COUPLING FACE

FIGURE 5.—KELVIN VIBROGRAPH

FIGURE 6.—GENERAL RADIO VIBRATION METER

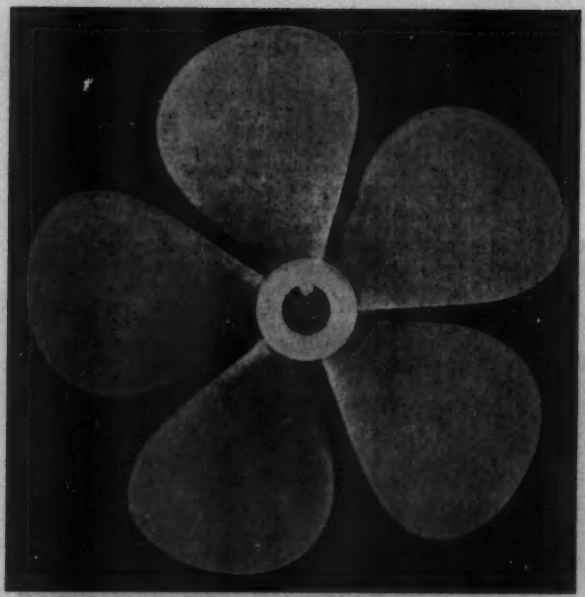

FIGURE 9(a).—FIVE-BLADED PROPELLER MODEL

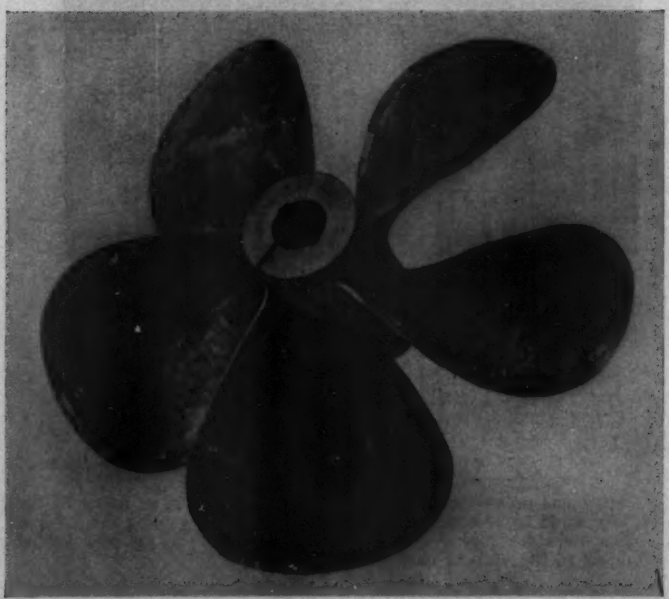

FIGURE 9(b).—FIVE-BLADED PROPELLER

wing and inner propellers. It is evident that if all propellers could be placed abreast, as in Figure 3, the trouble could not arise, but such an arrangement

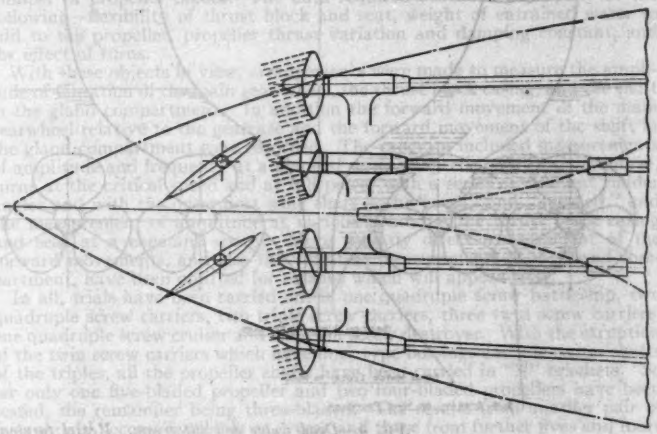

FIG. 3.—*Propellers all abreast.*

would be most undesirable in a warship from the point of view of action damage. For this reason the only possible improvement appeared to lie in increasing the distance and in more recent ships this is of the order of 40 to 50 feet.

The author's attention was first drawn to the problem in 1943, during discussion with another department regarding an apparent "critical speed" in H.M.S. *Furious* at about two-thirds of full power revolutions.

Calculations were made of the natural frequencies of torsional lateral and longitudinal vibration and the latter seemed the most probable, although it was necessary to make an unexpectedly large allowance for flexibility in the thrust block and seat in order to match the observed speed. Subsequent enquiry revealed that the thrust blocks of the inner shafts did in fact become extremely lively at the speed in question and that loosening of bolts, keys and rivets was a regular occurrence.

At this stage the records from *Warspite* were re-examined, and the longitudinal natural frequency was calculated, making similar allowances for thrust block flexibility. The calculations gave a natural frequency of about 900 cycles per minute, corresponding to a critical speed of 300 Rpm., i.e. full power, with 3-bladed propellers. From these results, it was concluded that resonance between propeller impulses and the natural frequency was largely responsible for building up high amplitudes of vibration. It was also concluded that the propeller thrust variation existed on straight course due to the blades passing through the comparatively dead water near the hull, and was much accentuated during turns if the blades also encountered, at the opposite end of a diameter, the fast running water in the slipstream from the wing screw, as shown in Figure 4.

Complaints were received at about this time from the *Illustrious* class, 3-shafted ships, regarding excessive vibration and difficulty in keeping packing in the center shaft stern gland. More recently these ships have suffered from loosening of stern tube bushes, loose rivets in thrust block seats and unduly rapid wear in the flexible couplings. Here again the speeds reported agreed with the calculated longitudinal critical of the center shaft.

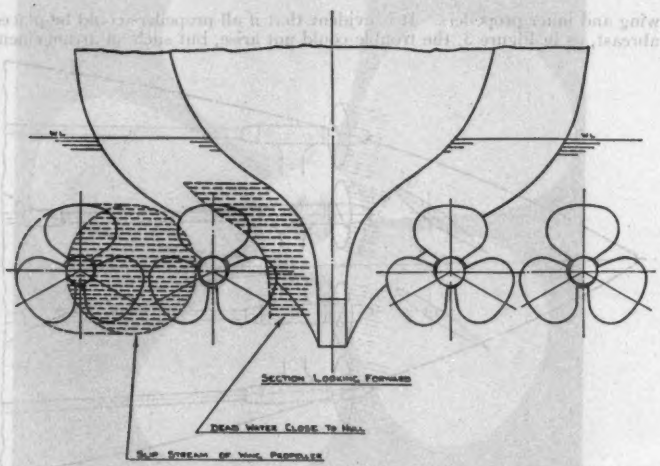

FIG. 4.—*Dead water and slipstream. Vessel turning to starboard.*

A further fillip was given to investigation by protests against the operational inconvenience of the easing procedure laid down for battleships, which made it necessary to sound the warning bell to the appropriate wing engine room before each turn, and also upset the operation of the boilers in that unit. Another less serious complaint concerned the resulting slight increase of turning circle diameter and time to turn.

THEORETICAL POSSIBILITIES OF IMPROVEMENT.

The conclusions so far reached, on admittedly slender evidence, suggested that resonance was an important factor in building up high amplitudes and that it might be possible to reduce the vibration to acceptable limits if it could be avoided. Two possible methods presented themselves: moving the thrust block further aft and so raising the critical above full speed, or increasing the number of propeller blades and running through the critical at a lower power where it might be harmless. The first method was attractive in that it avoided any loss of efficiency which might arise from a propeller with more than three blades, but could not readily be applied to existing ships. It was approved in certain new construction, but unfortunately the vessels concerned are among those cancelled at the close of the war. The possibilities and limitations of such a re-arrangement are discussed more fully in a later section of the paper. The second method was obviously the simplest for ships in service and appeared promising provided that such a propeller could be made to work without excessive cavitation.

SEA TRIALS.

It was obviously desirable that further experimental evidence should be collected before embarking upon the design and manufacture of a propeller, and arrangements were accordingly made to carry out vibration measurements in H.M.S. *Formidable*, with a view to ordering a five-bladed propeller for the center shaft if the results confirmed the importance of resonance. Similar trials were also arranged for new aircraft carriers then nearing completion

and it was hoped that sufficient fundamental data would be collected to allow reliable prediction of the performance of any arrangement of shafting and number of propeller blades. The data required consisted principally of the following—flexibility of thrust block and seat, weight of entrained water to add to the propeller, propeller thrust variation and damping constant, and the effect of turns.

With these objects in view, arrangements were made to measure the amplitude of vibration of the main gear wheel, the thrust block casing, and the shaft in the gland compartment. In addition the forward movement of the main gearwheel relative to the gearcase and the forward movement of the shaft in the gland compartment were recorded. The program included measurements of amplitude and frequency at a planned series of speeds on a straight course, turns at the critical speed and at full power with a series of different rudder angles and with the outermost wing shaft both "eased" and "uneased," and the measurement of amplitude at various points on the thrust block casing and seat at a constant speed. More recently direct measurement of the forward movements, and also the vibration measurements in the gland compartment, have been omitted for reasons which will appear later.

In all, trials have been carried out in one quadruple screw battleship, two quadruple screw carriers, two triple screw carriers, three twin screw carriers, one quadruple screw cruiser and one twin screw destroyer. With the exception of the twin screw carriers which have liner type bossings and the center shafts of the triples, all the propeller shafts have been carried in "A" brackets. So far only one five-bladed propeller and two four-bladed propellers have been tested, the remainder being three-bladed. The results from another pair of fives should become available next year and those from further fives and fours in two or three years time.

Instruments.

The particular range of frequency concerned, from 300 to 1250 vibrations per minute, is not over well provided for in respect of measuring instruments. For instance, the inductive type electrical pickups of the Sperry M.I.T. set, which has proved such a valuable tool in investigating vibrations in marine reduction gears, do not respond satisfactorily to such low frequencies.

This difficulty has led to the use of three entirely distinct types of instrument. The first includes the Cambridge Vibrograph and the Kelvin Vibrograph, both of which are hand held semi-seismic instruments which produce a scratched record, the former on a celluloid strip and the latter on pink waxed paper. Each automatically time marks the record, the former from a separate clock and the latter in a less constant manner from the rotation of its own paper drum. In each the scratching stylus is connected to a spring loaded probe which is held against the vibrating object with sufficient pressure to bring the stylus to a central position on the strip. Both instruments are satisfactory for their part of the job which is to provide a visible record from which frequency and shaft speed can be checked relative to each other. They are not altogether satisfactory for amplitude measurements because although the human body is an excellent damper, the calibration does appear to vary with different individuals and perhaps not surprisingly to be affected at times by the fact that the operator is standing on a vibrating base. Neither is convenient for use on a rotating shaft, such as the forward end of the gearwheel, because the spring pressure is not sufficient to hold a push rod in firm contact.

The second type of instrument used is the General Radio Vibration Meter, this has a crystal pickup contained in a case which can be held firmly against a push rod bearing on the forward end of the gearwheel shaft. The crystal pickup responds to acceleration and the resulting small electric impulses are amplified and integrated twice to give an r.m.s. vibration amplitude reading on a galvanometer. The instrument does not in the ordinary way give any indication of frequency, being designed for use with a low frequency analyzer

which has not yet become available in this country, but it can by the operation of selector switches read acceleration or velocity instead of amplitude and, for a purely sinusoidal vibration, frequency can be deduced from the relative values of these quantities. It has, however, been used during these trials purely to measure amplitude and with the exception of *Warspite* is responsible for all the amplitude figures quoted. Very great patience is required of the operator because the instrument has annoyingly unstable tendencies and he must watch and wait until he considers it has settled down before taking a reading. Similar patience, a steady hand and physical endurance are required of the individual who holds the pickup, often in a position of acute discomfort. An important point to note is that despite rubber feet the instrument picks up vibration through its case and must, therefore, always be held on the operator's knee. This point is illustrated in Figure 6 which shows calibration in progress, a check which is applied before and after every trial.

The third instrument used was a vernier gauge made for the purpose of measuring amplitude in the gland compartment. A section of the shaft was painted dull black and a circumferential line scribed through the paint to the bright surface while turning slowly and in the absence of vibration. When longitudinal vibration occurred the fine bright line expanded into a broader bright band and the vernier gauge, mounted on a bracket close to the shaft, was used to measure the width of the band.

The forward movement of the main gearwheel relative to the gearcase was measured with feelers and block gauges. A hole was drilled through the forward end cover in way of the end of the shaft, and a push rod inserted and held against it as shown in Figure 7. Measurements were made between the head of the pin and the outside of the end cover.

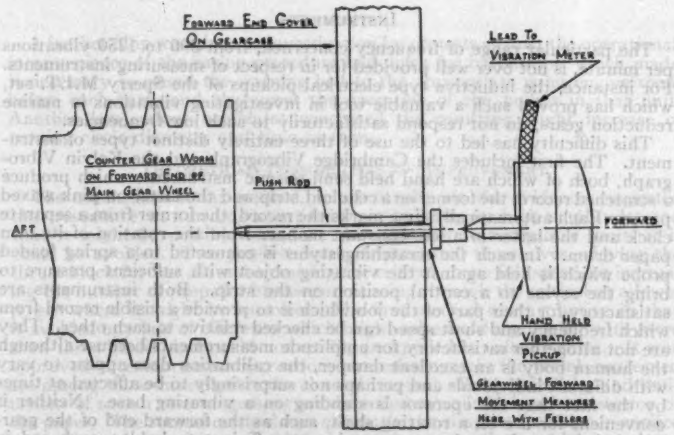
Forward End Cover On Gearcase
Lead To Vibration Meter
Counter Gear Worm on Forward End of Main Gear Wheel
Push Rod
Aft
Forward
Hand Held Vibration Pickup
Gearwheel Forward Movement Measured Here With Feelers

FIG. 7.—*Push rod and vibration pickup.*

RESULTS OF SEA TRIALS.

The readings taken during the first sea trials, on an inner shaft of a quadruple screw carrier, showed clearly that there was in fact a longitudinal critical speed, in this case slightly below full power, and that the frequency of vibration was as expected equal to three times the speed of revolution. A plot of vibration amplitude at the thrust block against Rpm. was found to

conform roughly to the shape of the resonance curve for a single degree of freedom system with a dynamic magnifier at resonance of ten. This meant that the propeller thrust variation, plus and minus some three tons, was being magnified tenfold by resonance to an alternating force of thirty tons, and confirmed the importance of resonance in building up large amplitudes.

The effect of turns was investigated at full power and the results obtained were generally similar to those from *Warspite*. It was found that when turning to starboard with equal power on all shafts the amplitude of vibration of the port inner was increased four to five times while for a similar turn with the port outer eased to 50 lb./sq. in. steam pressure the multiplier was only two.

The multiplier of four, in conjunction with the dynamic magnifier of ten, was increasing the three ton thrust variation to the considerable figure of ∓120 tons, approximately equal to the full thrust load.

The trials in H.M.S. *Formidable* followed some three months later and the results were generally similar except that the amplitudes on straight course were somewhat greater, as might be expected for a center propeller behind structure, and the effect of turns rather less. In this case the critical speed was lower and turns were carried out both at full power and at the critical with peculiar and misleading results, in that the amplitude of vibration was found to be the same in both cases. This led the author to conclude that the effect of turning increased very rapidly with speed and later to hope that a new ship with a critical well below full power would prove satisfactory with three-bladed propellers, an illusion rudely shattered by her first sea trials. It is now considered that the similarity of amplitude in the two cases must be attributed either to the rapid change of shaft speed occurring during turns, or to the breakdown of the vibrating system when thrust reversal occurs. The maximum amplitude observed in most ships has been approximately that required to cause the pressure on the thrust pads to vary from zero to double the steady thrust, but the vessel mentioned above proved a complete excep-

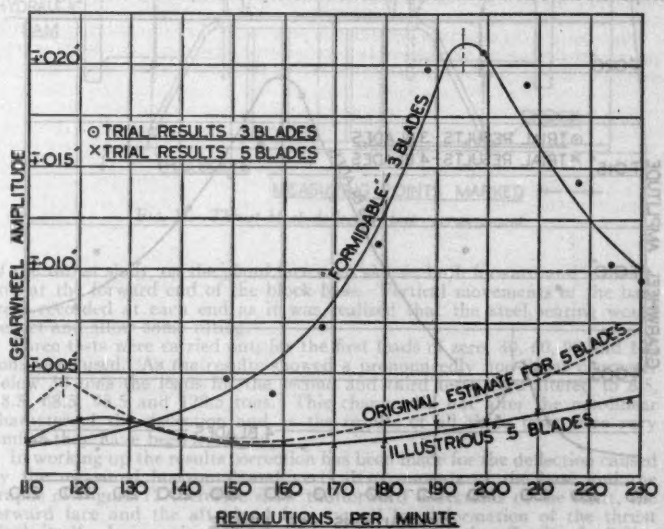

FIG. 8.—*Trial results—three- and five-bladed propellers.*

tion to this rule: the thrust collar jumped the full axial clearance between ahead and astern pads, and the pinions were thrown the full axial clearance of their teeth, moving over one quarter of an inch.

Figure 8 shows the amplitude of vibration of the main gearwheel in *Formidable* plotted against Rpm., and the dotted curve below is the prediction made at that time for the probable performance of a five-bladed propeller. It should be appreciated that somewhere to the right there is, theoretically at any rate, an alarming second critical whose position depends on the very elusive relationship between thrust block flexibility and propeller entrained water. Fear of this uncertainly placed second critical would have led to the choice of four rather than five blades, but it was considered that the former would give an even greater thrust variation than three blades if used behind the "sternpost" structure in way of this center shaft.

The five-bladed propeller shown in Figure 9 was in fact tried in H.M.S. *Illustrious* (a sister ship of *Formidable*) two years later and it will be seen that the results were even better than the original prediction. The improvement when turning was equally striking and the movement of the thrust block is now barely perceptible under the worst conditions.

Similar improvement has been found on fitting four-bladed propellers to the twin screw carriers. These vessels have liner type bossings instead of the usual Naval "A" brackets, and it was anticipated that this feature would increase the propeller thrust variation. Trials were accordingly arranged in an early ship of the class and as expected a critical speed was found in the long starboard shaft. The shorter port shaft proved to have its critical above full power. As might be expected the effect of turning was not serious with twin screws and it was not expected that the amplitudes observed would cause any damage. There was, however, considerable vibration of the hull and bridge structure at the critical speed of the starboard shaft and partly for this reason two starboard four-bladed propellers were ordered. These propellers have now been fitted with satisfactory results, as shown in Figure 10.

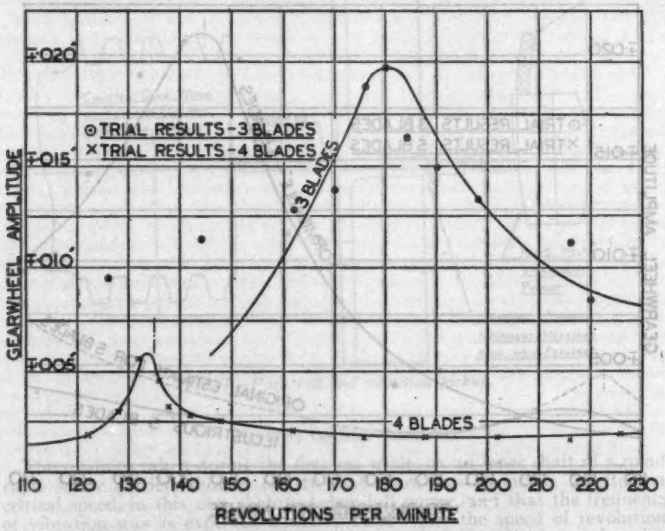

FIG. 10.—*Trial results—three- and four-bladed propellers.*

The trials in the four shafted cruiser and the twin screw destroyer were arranged with the object of finding out whether it was necessary to give any thought to longitudinal vibration in new construction ships of those classes. It was concluded that longitudinal criticals should be avoided by design in the former but might be accepted in the latter, provided that the critical speed was well below full power.

Other trials have yielded more comprehensive data on various aspects of the problem, including the effect of speed and rudder angles on the multiplier for turns, which is discussed in more detail in a latter section of the paper.

Deflection Test on a Thrust Block.

The first sea trials indicated that the thrust block itself, apart from the seating, was surprisingly flexible, and it was therefore decided to carry out a static deflection test on shore. This was done by Messrs. John Brown & Co., Ltd., using a thrust block belonging to a ship then under construction. A steel seating was built to carry the block and a hydraulic ram which was arranged to push on the aft end of the thrust shaft. Reference points for measurement were provided by brackets supported independently from the shop floor as shown in Figure 11. Forward movements were recorded at the forward end

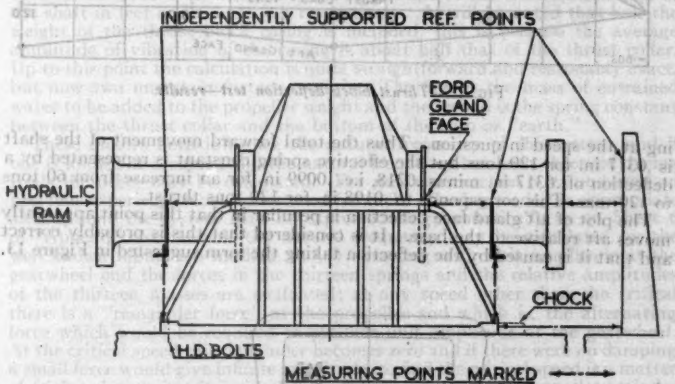

Fig. 11.—*Thrust block deflection test—arrangement.*

of the thrust shaft, on the gland face each side at both forward and aft ends, and at the forward end of the block base. Vertical movements of the base were recorded at each end as it was realized that the steel seating would deflect and allow some tilting.

Three tests were carried out, for the first loads of zero, 30, 60, 90 and 120 tons were used. As the results showed a pronouncedly non-linear character below 30 tons the loads for the second and third tests were altered to 8.5, 38.5, 68.5, 98.5 and 128.5 tons. This change did not alter the non-linear character of the deflection and as the results of all three tests were very similar they have been averaged.

In working up the results correction has been made for the deflection caused by the measured horizontal and vertical movements of the base and the graphs of Figure 12 therefore show the forward deflections of the shaft, the forward face and the aft gland face caused by deformation of the thrust block itself. It is evident that for vibration purposes the effective flexibility is not the total deflection but the slope of the line at the average thrust obtain-

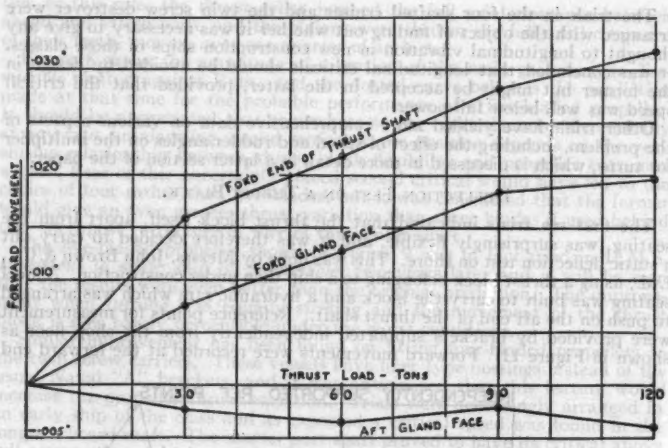

FIG. 12.—*Thrust block deflection test—results.*

ing at the speed in question. Thus the total forward movement of the shaft is .0317 in. for 120 tons but the effective spring constant is represented by a deflection of .0317 in. minus .0218, i.e. .0099 in. for an increase from 60 tons to 120 tons. This corresponds to .0198 in. for 120 tons thrust.

The plot of aft gland face deflection is peculiar in that this point apparently moves aft relative to the base. It is considered that this is probably correct and that it is caused by the deflection taking the form suggested in Figure 13.

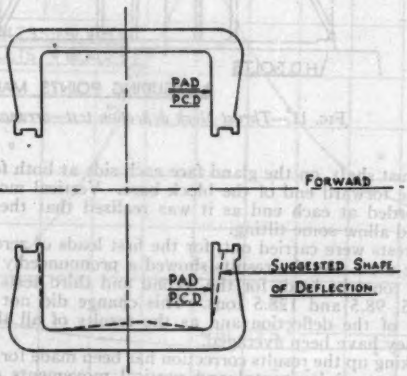

FIG. 13.—*Thrust block plan at cover joint*

If this deduction is correct it is evident that the thrust block could be made stiffer by thickening up the top flange and providing horizontal ribs. Deflec-

tion due to tilting of the base could be reduced by lengthening it (there is usually room for a base twice as long) and fitting a thicker top plate to the seat welded direct to the vertical members.

Methods of Calculation.

It is necessary at this stage to give an account of the standard method of calculation used in estimating critical speeds and amplitudes of vibration and in working up trial results. The method is set out in detail in the Appendices for the benefit of any prospective users and a brief explanation will suffice. The first step is to calculate or otherwise learn the weight of all the parts from the pinions to the propeller and to reduce the shafting to equivalent lengths of chosen cross section area. It is usually convenient to use two section areas of shaft by reason of the corrosion allowance on the outboard portion.

The system is then split up into (say) thirteen sections, the weight of each section is found, and the spring constant from the center of each section to the center of the next is calculated. This spring constant is the force in tons which would be required to increase the distance between the two points by one inch, and taking Young's modulus as 30×10^6 lb./sq. in. it amounts to $106.5 \times 10^3 \times A \div L$, where L is the length and A the cross section area of the shaft in feet and square feet respectively. It will be noted that half the weight of the thrust block casing is included, this is because the average amplitude of vibration of the casing is about half that of the thrust collar. Up to this point the calculation is quite straightforward and reasonably exact, but now two uncertain factors come in. The first is the mass of entrained water to be added to the propeller weight and the second is the spring constant between the thrust collar and the bottom of the ship or "earth."

These factors both influence the natural frequency of the system, and if one is known or assumed and the natural frequency has been observed during trials the other may be calculated. For new construction a value based on previous experience must be assumed for each. Having made the appropriate assumptions the system may be expressed diagrammatically as on Sheet 6 of *Appendix No. 1, and the natural frequency may be calculated by the well known tabulation method. [1, 2, 3] Starting with unit amplitude at the gearwheel end the forces in the thirteen springs and the relative amplitudes of the thirteen masses are evaluated; at any speed other than the critical there is a "remainder force" at the propeller end which is, the alternating force which would be required to produce unit amplitude at the gearwheel. At the critical speed this remainder becomes zero and if there were no damping a small force would give infinite amplitude, to find the critical speed is a matter of trial and error but it is usually possible to achieve a very small remainder at the third speed tried as the variation of remainder with speed is reasonably linear.

To estimate the amplitude of vibration at the critical two more assumptions, again based on previous experience, are required. These are the magnitude of the thrust variation, and the propeller damping factor. Dealing with thrust variation the important assumption is made throughout that the steady thrust varies as the square of the speed of revolution and that for a given propeller and hull the variation is a constant percentage of the steady thrust. This assumption appears to fit the results so far available.

Having selected suitable values for these factors the propeller amplitude at critical may be calculated from the simple relation (see ref. 1, page 66).

$$\text{Propeller amplitude} = P/\omega \times C_D$$

where amplitude = inches

P = alternating force, tons

ω = (Rpm.) $\times$ no. of blades $\times 2\pi/60$ radians per second.

C_D = propeller damping factor, tons per inch per second.

* Page 15 of I. M. E. Transactions, 1948, Vol. LX, No. 5.

This applies only at the critical speed.

The gear amplitude can then be found by the relationship already established in finding the natural frequency.

Amplitudes at speeds other than the critical may be calculated by the tabulation method; as unit amplitude is assumed for the gear its actual amplitude neglecting damping is simply

gear amplitude (inches) = P/remainder force.

Damping has no appreciable effect on the amplitude outside the range 0.8 to 1.2 times the critical speed. For a conventional line of shafting with the thrust block near the gearwheel end it is permissible to use standard resonance curves for a single degree of freedom to compute propeller amplitudes at speeds below and up to 1.3 times the first critical. This method saves a considerable amount of work and is particularly convenient in fitting a suitable damped resonance curve to trial results. The curves in Figure 14 have been computed

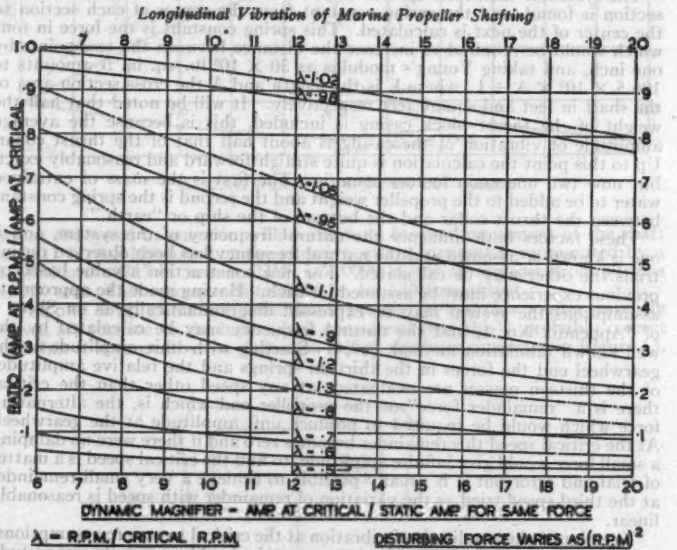

FIG. 14.—*Damped quadratic resonance curves.*

by the author for the purpose. The dynamic magnifier at the propeller is the calculated amplitude at the critical divided by the forward movement which would result if the same force were applied statically. It should be noted that the resonance curves cannot be used if the thrust block has been placed further aft because the system then becomes much more sharply tuned and its response is not adequately represented by a single degree of freedom. This limitation is illustrated in Appendix No. 2. The same applies to the second critical even in a conventional arrangement.

SPRING CONSTANT OF THRUST BLOCK AND SEAT AND PROPELLER ENTRAINED WATER.

During the first three trials efforts were made to measure the forward movement of the main gear wheel relative to the gear case, and to use the

measured movement in conjunction with the calculable propeller thrust to determine the spring constant.

The most serious difficulty lay in the fact that the gearwheel was vibrating with an amplitude of some ∓.010 inches at full power and a further complication was introduced by the expansion of the shaft between the thrust collar and the forward end of the wheel during the appreciable time required for working up. Despite these difficulties quite reasonable results were secured in the first ship, possibly due to special skill on the part of the individual making the measurements but probably largely attributable to good fortune. Results from later ships were not reasonable and the method was therefore abandoned.

The second method of evaluation, also used in the first three ships, was the measurement of amplitude in the gland compartment by means of the vernier gauge. It was hoped that from the relationship between gland compartment, thrust block and gearwheel amplitudes and the directly calculable spring constant of the shafting it would be possible to solve for the unknown spring constant of the thrust block. This proved unsatisfactory because of the different character of the measuring instruments, the G.R. Vibration meter recording a mean value over some seconds while the vernier gauge gave the maximum peak values. Synchronization of readings also offered difficulty, but probably this method could be made to work if sufficient attention were given to detail.

A third method which the author has not tried is the use of strain gauges to measure the alternating compressive stress in the shaft close to the thrust block, such measurements in conjunction with the vibration amplitude would give the spring constant directly. The compressive stress in a shaft is however only about 1,500 lb./sq. in. under full power thrust and by the time slip ring difficulties are taken into account the accuracy of measurement would probably be low. The method on which the present assumptions are based arises from the static deflection test previously described. The considerable movement relative to each other of the forward and aft gland faces of the block under the static load has been used as a means of calculating the alternating load from vibration amplitudes recorded at the same two points during subsequent sea trials. The wing shafts of this ship are longer than the inner and so give a lower critical speed, and if the same value of entrained water is allotted to each propeller it is found that to give the observed criticals the wing thrust block must be stiffer than the inner. This is confirmed by the above calculation which gives a flexibility, expressed as forward movement under full power thrust, of .036 in. for the wing and .042 in. for the inner shaft. Using these figures the weights of entrained water required to give the observed criticals agree within 2 per cent and the mean value of .0481 tons per square foot of developed blade surface has been adopted as a standard.

It should be noted that for a given first critical speed the response of the system to other frequencies does not show any marked change if entrained water is increased and thrust block flexibility reduced or vice versa until quite close to the second critical speed. The position of the latter is, however, controlled principally by the thrust block flexibility and when fitting a five-bladed propeller to a long shaft it is important to know that it will be above full power. The most satisfactory method of sorting out the true values of entrained water and flexibility would be to take the system up to the second critical with a vibration generator. It is considered that the friction in the shaft bearings would be too great if this were attempted in dry dock but consideration is being given to the possibility of attaching the vibration generator to the thrust block and carrying out the experiment while the ship is steaming at a moderate speed.

In the meantime trial results are worked up by using the above mentioned standard allowance of entrained water and calculating the thrust block flexibility from the observed critical speed. The resulting flexibilities vary in different ships from .036 inches to .057 inches forward deflection under full

DEN HARTOG[1] PAGE 62 GIVES THE FOLLOWING EQUATION FOR A SYSTEM WITH ONE DEGREE OF FREEDOM:-

$$x_0 = \frac{\frac{P_0}{k}}{\sqrt{\left(1-\frac{\omega^2}{\omega_n^2}\right)^2 + \left(2.\frac{c}{c_c}.\frac{\omega}{\omega_n}\right)^2}}$$

WHERE P_0 = A CONSTANT APPLIED ALTERNATING FORCE OF THE FORM $P = P_0 \sin \omega t$

ω = THE APPLIED FREQUENCY, RADIANS/SEC.

ω_n = " NATURAL " , " "

k = THE SPRING CONSTANT.

c = THE DAMPING FACTOR.

c_c = THE CRITICAL DAMPING FACTOR

x_0 = THE RESULTING AMPLITUDE OF VIBRATION

SUBSTITUTING $\lambda = \frac{\omega}{\omega_n}$

AND $D = x_{0\lambda} \Big/ \frac{P_0}{k}$ WHERE $x_{0\lambda}$ = AMPLITUDE FOR $\lambda = 1{\cdot}0$

THIS MAY BE TRANSPOSED TO THE FORM:-

$$R_\lambda = \frac{1}{\sqrt{D^2\left(1-\lambda^2\right)^2 + \lambda^2}}$$

WHERE R_λ = AMPLITUDE AT FREQUENCY $\omega = \lambda\omega_n$ / AMPLITUDE AT FREQUENCY ω_n

AND D = THE DYNAMIC MAGNIFIER AT RESONANCE

THIS EQUATION IS FOR A CONSTANT VALUE OF P_0 WHICH FOR THE THE PRESENT PURPOSE IS IN FACT TO BE PROPORTIONAL TO $(R.P.M.)^2$ AND SO TO λ^2.

THE FINAL FORM USED FOR PLOTTING FIG. 14 THEN BECOMES:-

$$R_\lambda = \frac{\lambda^2}{\sqrt{D^2\left(1-\lambda^2\right)^2 + \lambda^2}}$$

FIG. 15.—*Derivation of formula for damped quadratic resonance curves.*

power thrust. In estimating the critical speed for a new ship with thrust block and seat of conventional proportions a value of .045 inches deflection under full power thrust should give results not far from the truth.

Propeller Thrust Variation.

The value of thrust variation deduced from trial results is dependent on the amount of entrained water assumed because that affects the calculation of thrust block flexibility and hence the force required to vibrate the gearwheel with the observed amplitude. For this reason the figures given below must not be divorced from their context: if at any future date the standard figure for entrained water is changed then the values of thrust variation must be re-calculated.

If a reliable portion of the resonance curve is available outside the range 0.8 to 1.2 times the critical the thrust variation may be calculated directly as the remainder in the tabulation method for the observed gearwheel amplitude and frequency. Unfortunately this is rarely possible both because at the small amplitudes then obtaining other vibrations are liable to add appreciably to the amplitude and because there is usually considerably scatter of the points. It has therefore been usual to find the dynamic magnifier at the propeller first and then calculate thrust variation from the amplitude at the critical.

To find the dynamic magnifier at the propeller from trial results is a curve fitting process. The ratio propeller amplitude/gear amplitude is first calculated by the tabulation method for say 0.7, 1.0 and 1.3 times the critical speed and a curve of this function is plotted. The propeller amplitude at the critical is found from the observed gear amplitude and the above ratio. Using this as the fixed point a series of resonance curves for the propeller motion are calculated from the curves of Figure 14 using different values of the dynamic magnifier. Each is converted to gear amplitude and the best fitting is adopted. By definition the thrust variation is the force which if applied steadily would move the propeller forward (by deflection of the thrust block and compression of the shaft) an amount equal to propeller amplitude at critical divided by the dynamic magnifier.

The results from the trials so far analyzed show that the following figures may be taken as representative of the propeller thrust variation with the ship on a straight course:—

3 blades with "A" brackets...	∓ 3 per cent variation.
3 blades with bossings.....	∓ 4 per cent variation.
3 blades center shaft.....	∓ 5 per cent variation.
4 blades.....	¼ of above values except for the center shaft where the variation would be greater than with 3 blades.
5 blades.....	3/5 of 3-blade values.

It should be noted that H.M.S. *Formidable*, the basis of the figure for a center shaft, has the structure sloped away below the shaft so that her stern is midway between a bossing and a sternpost. It is probable therefore that a single screw merchant ship would give a considerably higher thrust variation, especially with a four-bladed propeller.

The figure of 3 per cent for three blades and "A" brackets is close to the observed value of propeller torque variation under similar conditions.(6)

Propeller Damping Factor.

It has been assumed throughout that all the damping present is at the propeller. This is certainly not quite true, but so long as the shaft and gearwheel are supported on water and oil films in their bearings the only other appreciable source of damping appears to lie in friction between the poorly lubricated teeth of the flexible couplings. This is an uncertain quantity depending on the condition of the surfaces; the turbine rotors do not partake of the axial motion unless the coupling teeth have become very rough when the

balanced double flow L.P. may do so. It does not appear therefore that the force is very great and when the thrust block is in the normal, forward, position it is quite satisfactory to ignore this source of damping. When however the thrust block is 200 feet from the gearwheel as in Case 2 of Appendix No. 3 it is just possible that the damping at the forward end might be sufficient to suppress the gear wheel motion and therefore invalidate the conclusions drawn regarding very long shafts. The best way to find out would be to fit an extra thrust block well aft on such a line of shafting and run trials with thrust pads fitted first in the aft block and then in the forward one.

Assuming that all the damping is at the propeller the damping factor may be found from the propeller amplitude at critical, thrust variation, and frequency, using the relation between these factors previously quoted. Up to the present the result has been expressed in terms of propeller developed surface, as for entrained water, and a representative figure is .00739 tons per inch per second per square foot of developed blade surface. The results of the trials with four- and five-bladed propellers suggest however that the damping factor increases with the number of blades, and if this is so possibly it would be better expressed as .0177 tons per inch per second per foot of blade edge length. This point is under investigation.

The Effect of Turns.

The effect of turns has been studied under various conditions of speed and rudder angle on an inner shaft of a quadruple screw ship and on the center shaft of H.M.S. *Illustrious* with the five-bladed propeller fitted. When a 4 shaft ship turns to starboard with full rudder the port inner normally suffers a violent burst of vibration soon after the swing starts, becomes quiet while the ship is doing a steady circle and then has another violent burst as she straightens up. The violent spells presumably occur when the slip stream of the wing propeller covers about half the disc of the inner, and the results from the quadruple screw ship show that there is a rudder angle less than the maximum which will cause violent vibration to be maintained throughout a circle. This does not apply to the center shaft of the triple because there the ship's structure prevents the slip stream from covering more than half the disc.

The results of the tests at different speeds show that over the range half to full revolutions there is little change in the effect. The "turn factor", amplitude on turn/amplitude on straight course, has maximum values in the quadruple screw ship of 4.96 at half speed and 5.4 at full speed while in *Illustrious* it is 4.1 at 102 Rpm., 4.7 at 170, 3.7 at 210 and 3.2 at 220 Rpm. These results are all for speeds where there is no reversal of thrust.

In estimating amplitudes for new construction a "turn factor" of 5 is used for quadruple screws in "A" brackets and a factor of 4 for the center shaft of a triple. The latter figure does not mean that the triple suffers less on turns, merely that it is worse on straight course.

The effect of turning is less in twin screw ships, the turn factor amounting to about 2 and affecting principally the shaft on the inside of the turn.

Apart from the increase of vibration another result of turning is important where fitting a five-bladed propeller may leave the second critical close above full power and where moving the thrust block aft may place the first critical in the same position. When a quadruple screw ship turns to starboard and the slip stream of the port wing shaft covers the inner propeller the latter is relieved of its thrust and speeds up rapidly. This increase of speed is liable to take it 10 per cent above normal full power revolutions and cannot be controlled when the tachometers fitted are of the integrating type. It is therefore necessary in the cases mentioned to consider revolutions up to 110 per cent of normal maximum.

Approximate Method of Finding Critical Speed.

When examining a proposed arrangement of shafting or drawing up a trial program for a ship it is desirable to have a short method of estimating the

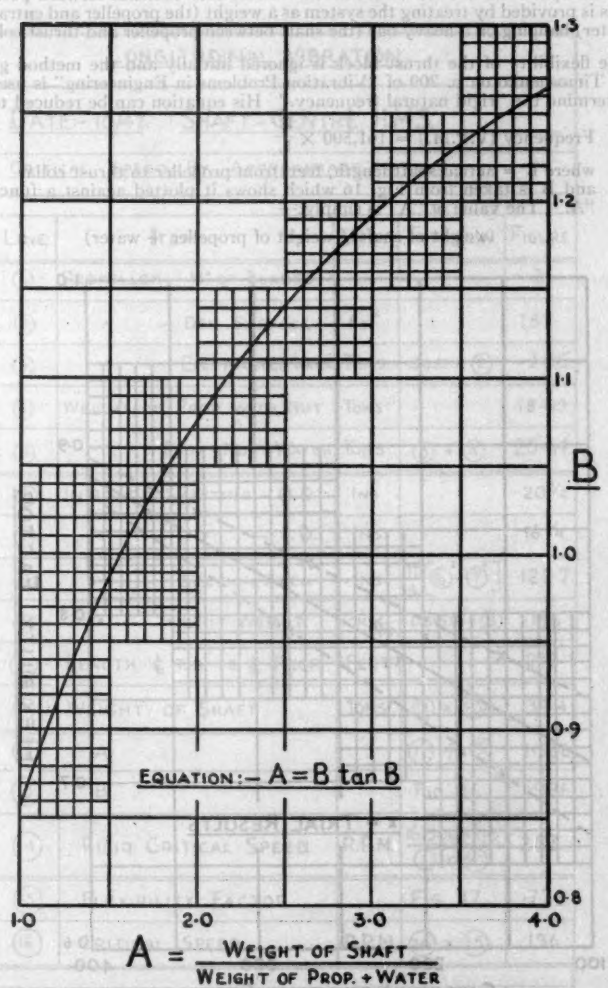

FIG. 16.—*Approximate method—graph of A and B.*

critical speed. Where the thrust block is in the conventional forward position this is provided by treating the system as a weight (the propeller and entrained water) hanging on a heavy bar (the shaft between propeller and thrust collar).

The flexibility of the thrust block is ignored initially and the method given by Timoshenko on p. 209 of "Vibration Problems in Engineering" is used to determine the "rigid natural frequency." His equation can be reduced to:—

$$\text{Frequency (V.P.M.)} = 161{,}500 \times \frac{B}{L}$$

where L = actual shaft length, feet, from propeller to thrust collar and B is taken from Fig. 16 which shows it plotted against a function "A." The value of "A" is simply:—

(weight of shaft)/(weight of propeller + water)

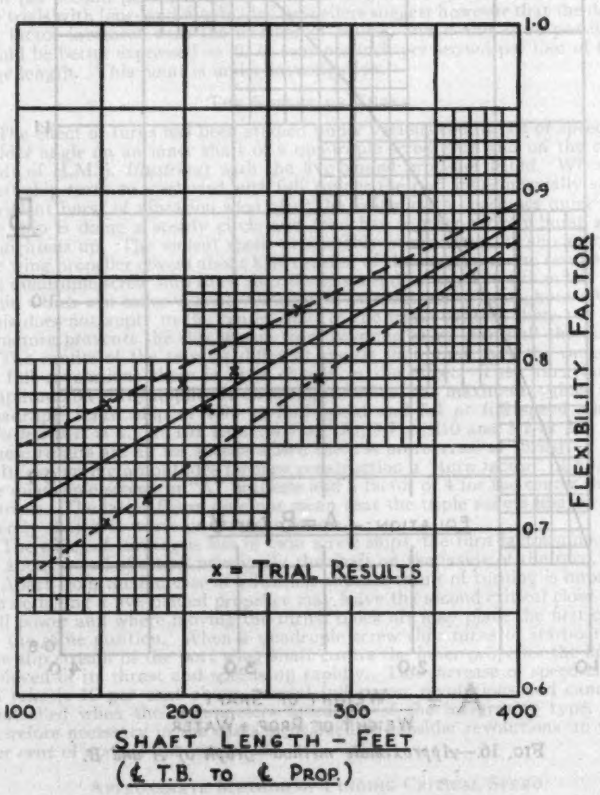

FIG. 17.—*Approximate method—flexibility factor.*

LONGITUDINAL VIBRATION

DATE:- 10/47 SHAFT:- CENTRE H.M.S. _____

CRITICAL SPEED BY APPROXIMATE METHOD

LINE	ITEM	UNITS	FORMULA	FIGURE
①	PROPELLER – Nº OF BLADES			3
②	" – DEV. SURFACE	$FT.^2$		157
③	" – ENTRAINED WATER	TONS	·0481 × ②	7·55
④	WEIGHT OF PROP. WITH NUT	TONS		18·09
⑤	" " PROP. + NUT + WATER	TONS	③ + ④	25·64
⑥	IN BOARD SHAFTING – O.D.	INS.		20½
⑦	" " – I.D.	INS.		16¼
⑧	" " SECTION AREA	INS^2	$\frac{\Pi}{4}[⑥^2-⑦^2]$	122·7
⑨	" " UNIT WEIGHT	TONS/FT.	·001516 × ⑧	·186
⑩	LENGTH ℄ T.B TO ℄ PROP.	FEET		212
⑪	WEIGHT OF SHAFT	TONS	⑨ × ⑩	39·4
⑫	A		⑪ ÷ ⑤	1·536
⑬	B		FIG. 16	·996
⑭	RIGID CRITICAL SPEED	R.P.M.	$\frac{161500 \times ⑬}{① \times ⑩}$	253
⑮	FLEXIBILITY FACTOR		FIG. 17	·775
⑯	CRITICAL SPEED	R.P.M.	⑭ × ⑮	196
	NORMAL FULL SPEED	R.P.M.		230

FIG. 18.—*Approximate method—worked example.*

and for this purpose the weight of the shaft is taken as the actual length from thrust collar to propeller multiplied by the weight per foot of the bare shaft. Couplings, gunmetal liners, etc., are ignored. The weight of entrained water used is .0481 tons per square foot of developed blade surface, as in the full method.

When the "rigid natural frequency" has been calculated it must be multiplied by a "flexiblity factor" to correct for the effect of thrust block flexibility. This has more influence on the natural frequency of a short shaft than on that of a long one, and Figure 17 gives a plot of flexibility factor against shaft length based on trial results to date. If thrust blocks and their seats are made more rigid this factor will increase towards unity.

Figure 18 shows an example worked by this method.

The Appendices.

Note: The appendices have been omitted since they are long. Those interested will find them in I. M. E. Transactions, 1948, Vol. LX, No. 3.

Appendix No. 1 gives the complete calculations for the natural frequency and amplitude of vibration of the center shaft as fitted in H.M.S.'s *Formidable* and *Illustrious* with three- and five-bladed propellers. The calculated amplitudes on straight course are reproduced in Figure 19 and it is seen that reason-

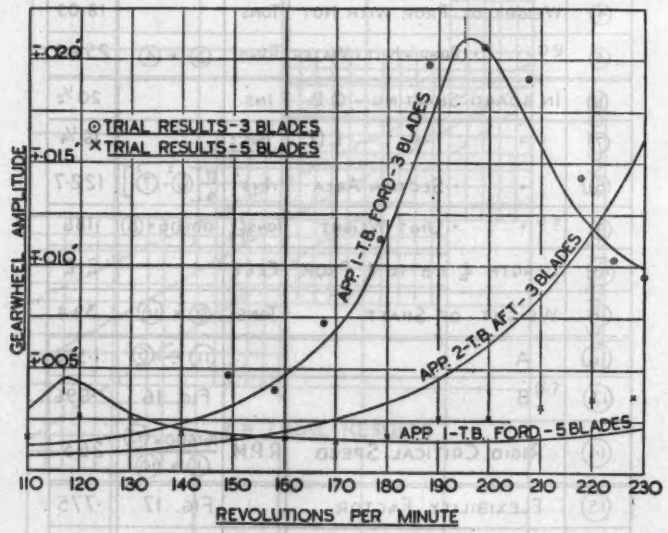
±.020"
±.015"
±.010"
±.005"
GEARWHEEL AMPLITUDE
⊙ TRIAL RESULTS - 3 BLADES
× TRIAL RESULTS - 5 BLADES
APP. 1-T.B. FORD - 3 BLADES
APP. 2-T.B. AFT - 3 BLADES
APP 1-T.B. FORD - 5 BLADES
110 120 130 140 150 160 170 180 190 200 210 220 230
REVOLUTIONS PER MINUTE

Fig. 19.—*Results from Appendices Nos. 1 and 2.*

able agreement with the trial results is secured. If Figure 19 is compared with Figure 8 it will be seen that the present calculated amplitudes with five blades fall considerably below the original estimate at the higher speeds. The reason for this is that the original estimate was made by calculating amplitudes and dynamic magnifiers at the first and second criticals and then filling in the space between by fairing together the appropriate single degree of freedom resonance curves, a procedure since found to be invalid for the flank of the second critical.

It will be noticed on Figure 19 that the trial results for the five-bladed propeller fall below the calculated values at the left and above them at the right. The former discrepancy suggests that the five-bladed propeller has a higher damping factor than the three-bladed one, a point already discussed. The increased amplitude at higher powers is probably due to the center shaft picking up from the hull the appreciable three per revolution vibration arising from the three-bladed wing propellers. This can be seen quite clearly on the Kelvin Vibrograph record at 200 Rpm. where in spite of the five-bladed propeller the center shaft breaks into free vibration at 600 Vpm. which is apparently the natural frequency in *Illustrious*. From this and the position of the first critical it is concluded that *Illustrious* has a slightly stiffer thrust block seat than *Formidable*.

Regarding turns the calculations show a heavy reversal at the critical speed with a three-bladed propeller which is confirmed by hammering in the thrust block and rattling of the gears, and a slight reversal at the critical with a five-bladed propeller which does not in fact occur.

At the critical speed, on straight course, the anticipated alternating force on the thrust block seat is reduced from ± 47 tons with three blades, to ± 10 tons with five blades, and the trial results show an even greater reduction.

Appendix No. 2 deals with the same line of shafting as Appendix No. 1, but the thrust block has been moved to a new position about mid-way between gearwheel and propeller and at the same time its stiffness has been increased from 2.27×10^3 to 3.25×10^3 tons/inch. The result of these alterations is to raise the critical speed with a three-bladed propeller from 195 to 250 Rpm. and as can be seen in Figure 19 this is a distinct improvement. The amplitude at full power is however still too great and if such an alteration were contemplated it would be necessary to increase the thrust block stiffness still further, to at least 4.0×10^3 tons/inch.

Appendix No. 3 deals with a hypothetical shaft 400 ft. long having the same scantlings and running at the same speed as those in Appendices 1 and 2. The gear, thrust block and propeller weights have been rounded up and, to save labor, couplings, gunmetal liners, etc., have been disregarded.

Three different thrust block positions have been considered and in each case the stiffness has been taken as 4.0×10^3 tons per inch. This figure is unusually high but would in the author's opinion be quite easy to achieve.

In Case 1 the thrust block is in the normal position 10 feet aft of the gearwheel and as would be expected with so long a shaft the first critical with three blades occurs at a fairly low power. The gearwheel amplitude is only $\pm .0058$ in. and on straight course the arrangement would be satisfactory except that the alternating force of ± 22.9 tons imposed on the thrust block seat might be sufficient to cause hull vibration in the event of resonance. When turning however there will certainly be a thrust reversal. If the thrust collar elects to jump the .050 in. clearance between ahead and astern pads and the pinions also shuttle the resulting amplitude will be sufficient to damage the flexible couplings. Such jumping and shuttling is probably more likely where the critical occurs at a low speed because the torque loads on coupling and gear teeth are reduced.

With five blades the first critical will be moved down to 88 Rpm. and there may just be a thrust reversal when turning. If however the five-bladed propeller shows increased damping as in *Illustrious* there will certainly be no reversal. Thanks to the specially stiff thrust block the second critical is still above full power and as shown in Figure 20 five blades will give a completely satisfactory performance throughout. It should be noted that the vertical scale of Figure 20 is one quarter that of Figure 19.

In Case 2 the thrust block is placed midway between gear and propeller. The amplitude of vibration at the gearwheel has been calculated neglecting any damping at that end and the results are also plotted on Figure 20. It will be seen that even this position of the thrust block (which gives the maxi-

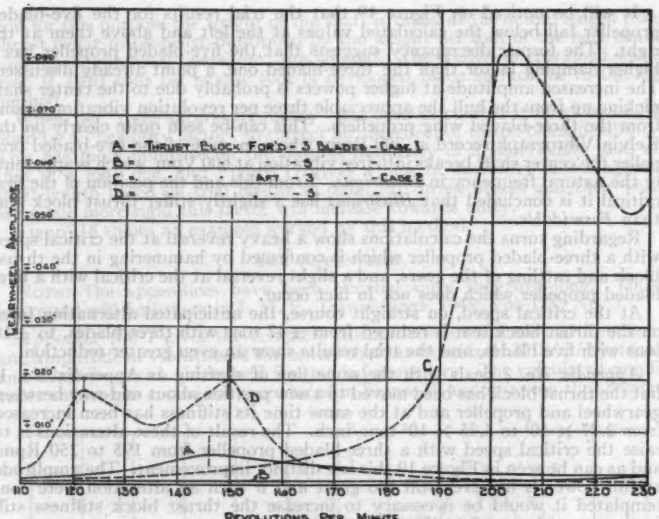

FIG. 20.—*Results from Appendix No. 3.*

mum natural frequency) fails to move the first critical with three blades above full power, and that the gearwheel amplitude is very great, principally because it is now equal to that of the propeller. The results with five blades show considerably reduced but still excessive amplitudes and illustrate the adverse effect of moving the thrust block on the second critical which has been brought down quite near to the first.

The enormous amplitudes resulting when damping at the gearwheel end is neglected have an air of unreality and are evidently too great. For this reason the force required to hold the gearwheel stationary has been calculated and it is found that with three blades at 226 Rpm. (the critical speed with the wheel held still) it would require a force of ± 10.1 tons on straight course. The Lp. turbine is double flow and therefore the only anchor available is the steam thrust of the Hp., amounting to 4.46 tons at full power. Assuming a co-efficient of friction of 0.1 for axial sliding in the flexible couplings the force available to hold the gear is 2.6 tons and it is evident that this will not be able to keep it stationary even on straight course. To calculate the actual gear amplitude under these conditions would be a trial and error process of assuming different amplitudes, until one was found at which the energy dissipated against viscous damping at the propeller and "dry friction" at the coupling equalled the energy put in by the thrust variation. A further complication would arise because the gear motion would not be truly harmonic.

Even with the gear held still the force on the thrust block seat is ± 52.3 tons on straight course at 226 Rpm. and for this reason the arrangement is in any case not acceptable with three blades. If five blades must be used there is no point in moving the thrust block at all.

In Case 3 the thrust block is placed only 80 feet from the propeller, as far aft as possible, and the force required to hold the gear stationary is again calculated. It is found that if the gear can be held the first critical will lie above full power and that the force required to hold it with three blades at

230 Rpm. is only ±1.72 tons. It appears probable that with this arrangement the gear would actually stay still on straight course but would move when turning. The force on the thrust block seat on straight course is only ±11.5 tons with the gear stationary.

It does seem possible that the arrangement of Case 3 might allow the use of three bladed propellers with satisfactory results, but before committing himself to such an arrangement the author would wish to secure measured values of the co-efficient of friction in flexible couplings and find a satisfactory method of predicting the resulting amplitudes.

Permissible Amplitudes.

The preceding pages have given the data and the methods for predicting the amplitude of vibration with normal shaft arrangements but have not given any indication as to what is considered acceptable. From the machinery aspect the principal criterion is excessive wear in the flexible couplings. Experience in Naval vessels shows that an amplitude of ±.020 in. on straight course is sufficient to cause rapid wear and that ±.050 in. when turning can cause heavy scoring and even seizure of the tooth surfaces. It is concluded therefore that the amplitude should not exceed ±.010 in. on straight course and ±.025 in. when turning.

In twin screw ships the ±.010 in. on straight course is the limiting factor, subject to the overruling consideration that a thrust reversal must not occur when turning. In triple or quadruple screw ships with a "turn factor" of four or five the amplitude on the turn is the criterion and to keep to the limit of ±.025 in. the amplitude on the straight should be kept down to ±.005 in. When the critical occurs at a low speed the avoidance of thrust reversal becomes more important than actual amplitude.

Single Screw Merchant Ships.

The author has had no practical experience with any single screw vessels except the American built ships converted to aircraft carriers during the war. In these there was no longitudinal vibration problem because even with four-bladed propellers the critical was well above full power. This is probably the situation in most single screw ships, though Den Hartog[1] has apparently met with a case where the critical came at full power. This case is used for his examples 69 and 70 on page 202 of "Mechanical Vibrations."

So long as there is no critical speed in the running range there is obviously no objection to using a four-bladed propeller behind a sternpost, in fact it is quite likely that a three-bladed propeller would give a heavy six per revolution disturbance in such a situation and so bring in a second order critical. The four-bladed propeller would be expected to give a very strong four per revolution variation but not much eight per revolution disturbance.

Precise Location of Thrust Block.

All the foregoing data and calculations have referred to thrust blocks which have been separate from the gearcases and supported on seatings which are virtually independent of those for the main machinery. This is a very important point, because if the thrust block is located in the gear case as for instance in the "locked train" design, the gearcase, turbines and condenser all become involved in the longitudinal vibration[6]. It is also most difficult to provide adequate fore and aft stiffness under a large gearcase because its own sump and the condenser are in the way. For this reason it is advisable where a high stiffness figure is required to sacrifice the possibility of saving space and keep the thrust block on a separate seat.

In all cases a stiff thrust block is helpful in reducing vibration amplitudes on straight course, but may increase the likelihood of thrust reversal when turning. The stiff thrust block in Appendix No. 3, Case 1, is ideal if a five-bladed propeller is to be used, but a more flexible one would perhaps reduce the severity of thrust reversal with three blades.

Increasing Shaft Flexibility.

Dr. Forsyth(4) has suggested that where the critical is close to full power it would be an improvement to introduce a flexible element such as a bellows just aft of the thrust block. This would move the critical to a lower speed and also reduce the ratio of gear to propeller amplitude. The detailed design and possible flexibility of such a device require to be worked out and the resulting amplitudes could then be calculated. The only possible objection to this proposal is that the amplitude of vibration of the bulk of the shaft, which is the greatest mass in the system, would be increased relative to the propeller amplitude. The idea is under investigation and until figures are available, no final conclusion can be reached as to its merits.

Multi Bladed Propellers.

It will have been noticed, with particular reference to Appendices Nos. 2 and 3, and to the preceding paragraph, that while the use of four or five blades instead of the standard naval figure of three is known to give entirely satisfactory results as regards vibration, there is a desire to explore other methods. The reason for this is that there is likely to be some loss of efficiency and increased risk of root cavitation when the number of blades is increased. The results so far do not make it possible to give any figures for efficiency but tend to suggest a slight loss, and as neither the five- nor the four-bladed propellers have yet been docked it is not known whether there has been any cavitation erosion.

General Principles.

A. Existing Ships.

Once a ship is complete it would be a major operation to move the thrust block and for this reason the only practical alterations are to change the number of propeller blades or to increase the shaft flexibility. If the critical is near full power and the shaft is in "A" brackets, there is a free choice between four blades and five blades. Five blades will give the greater certainty of smooth running and the avoidance of thrust reversal on turns, but a four-bladed propeller is likely to be slightly more efficient and less prone to cavitation troubles. Where the critical is some way below full power with three blades it is possible that the use of five will bring the second critical into the running range and it is wise to calculate the probable position of this.

Where the propeller works behind a sternpost and the critical is within the running range, four blades should not be used.

It may occasionally happen that the first critical is at, or just above full power. In such a case it is worth considering the possibility of fitting a new and stiffer thrust block in the original position and at the same time stiffening the seat. In many cases it would be possible to fit a block with a base twice as long on the existing seat.

B. New Construction.

Shafts may be divided, having regard to their running speed, into four classes: short, medium, long and very long.

With a short shaft, the first critical is above full power with three blades and no special precautions are required. Slightly longer shafts may be brought into this category by making a special effort to provide a stiff thrust block and seat.

With a medium length shaft the first critical comes below full power with three blades when the thrust block is near the gearwheel end, but may be raised well above full power by placing the thrust block midway between gearwheel and propeller. In this case the thrust block can be so located and a three-bladed propeller may then be retained. This arrangement requires either a self-contained, self-lubricating thrust block, or a separate lubricating oil system, because the block is too far aft to allow oil to drain back to the

engine room. Allowance must also be made at the gears for expansion of the shaft between thrust block and gearwheel. This will depend on the number of plummer blocks and bulkhead glands fitted; measurements taken in one ship show a movement relative to the hull of about 5/16 in. per hundred feet.

A long shaft is defined as one in which the first critical cannot be moved above full power by moving the thrust block half way to the propeller. In this case it becomes necessary to change to four- or five-bladed propellers with the possible alternative of increasing the shaft flexibility, as discussed under "Existing Ships." The thrust block should be kept in the forward position and made sufficiently stiff to ensure that the second critical is well above full power with the chosen number of blades.

With a very long shaft it appears possible, if the thrust block is placed as near as practicable to the propeller, for the friction in the flexible couplings to hold the gearwheel stationary under all conditions. If the truth of this conclusion can be established, and if the retention of three-bladed propellers is regarded as very important, this arrangement will warrant serious consideration.

SUMMARY.

Longitudinal vibration can be reduced to acceptable amplitudes by fitting a propeller with an increased number of blades but this may involve some sacrifice of propulsive efficiency. The use of three-bladed propellers may be extended to shafts of medium length by locating the thrust block midway between gearwheel and propeller. In long shafts such a modification would fail to raise the critical speed above full power and it is best to keep the thrust block in the normal position and use either more propeller blades or perhaps a device to increase shaft flexibility. With very long shafts it appears possible that friction at the flexible couplings may be able to hold the gearwheel stationary if the thrust block is placed as near as possible to the propeller.

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THE EFFECT OF NON-CONTACT EXPLOSIONS ON WARSHIP MACHINERY DESIGN.

This paper by Commander (E) A. D. Bonny, R. N., is reprinted from a copy of the paper issued by the Institution of Naval Architects prior to its presentation on March 18, 1948. It deals with a subject of great importance to naval engineers and one which is the subject of much study today. Photographs of typical damage, Figures 2 to 11 incl., 14, 15, and 16, have been omitted.

Summary.

The widespread damage to machinery caused by non-contact under-water explosions caused much concern early in the war, and several trials have been carried out to investigate this problem. Photographs of typical damage on service have been included to show the serious nature of the problem.

The phenomena caused by an under-water explosion are briefly described, and the effect of the pressure pulse caused by it traced to the ship and thence to specific machinery items. While it is not possible to formulate a definite theory to cover the motion caused by the explosion, it is found that a close approximation can be made to the velocity of an item in a ship from energy principles, a simplified theoretical basis for this being given in Appendix I.

The magnitude of the accelerations are then discussed and the effect of such suddenly applied motion on machinery design is considered.

Since one of the first requirements was to modify existing machinery to withstand these shock forces as far as possible details of typical modifications are given in Appendix II.

As shock is not the only criterion of design, many machines cannot be designed to withstand such forces, and special mountings to reduce the forces are discussed.

Introduction.

Before the 1939–45 war the problem of damage to machinery due to shock had not been one of importance; the machinery was in general of a robust nature, as was essential for the conditions of naval service, and had withstood, without any outstanding troubles, the vibration met with at high speeds and also the shocks due to the firing of guns and the dropping of depth charges. Previous war experience had not revealed any weakness in this respect when ships were hit by shells or torpedoes, though the possibility of transmission of shock through the water had been considered in so far as it might affect condenser inlets and doors. Some damage from blast had also occurred when ship's guns were trained too close to the ship's structure.

Late in 1939, however, H.M.S. *Belfast* was mined in the Firth of Forth by a magnetic mine on the sea-bed, and the very extensive damage to machinery and hull structure caused by this mine exploding some 80 to 90 ft. away from the hull focussed attention on this new type of attack.

This incident was followed by many more of a similar nature when the various types of non-contact mines employed by the enemy, as well as the explosion under water of near-miss bombs, caused a considerable number of ships to be laid up for long periods of repair. In all, over 100 ships suffered shock damage during the war—much of it of a serious nature.

Since, in general, warships are specially designed to withstand attack by torpedoes, mines, projectiles, bombs, and any other method of attack, this result was unexpected and formed a new and major problem for the naval designer at a critical period of the war.

An examination of the problem showed that in most cases the machinery and electrical equipment suffered greater damage than the hull itself, except in those cases where the ship suffered partial collapse of the structural girder. In the case of the machinery and equipment, by far the greatest part of the damage was caused by fracture of cast-iron castings, though considerable subsidiary damage and derangement was caused by the tripping of governors and defects in electrical breakers, lighting, and other electrical equipment, gyro compasses in particular suffering severe damage.

It is especially noteworthy that very little of this shock damage occurs from direct hits by either shell or bomb or from torpedo hits. In these cases the local devastation is severe, but once outside the direct radius of the explosion little damage other than by splinters is met.

Though the incidence of shock damage fell off after 1941–42, due to the various protective measures against magnetic and acoustic mines, the necessity for design against shock is still of great importance.

Since most of the design departments were affected by this problem, an inter-departmental committee, the Admiralty Shock Committee, was formed to investigate the problem both in theory and by experiment. This committee has carried out full-scale experiments on H.M.S. *Cameron*, a Town class destroyer; on the submarine H.M.S. *Proteus;* on a target representing a section of a submarine; and on the destroyer H.M.S. *Ambuscade*, while further investigations into the effect on larger ships are being pursued.

Without going into detail of these experiments, the following general picture of the effects of an under-water explosion will be of value in formulating principles for the design of machinery to withstand shock.

Description of an Under-Water Explosion.

Taking as an example a medium-size charge of 300 lb. T.N.T. (or its equivalent of other explosive) in a container under water, then, on detonation, the solid explosive is changed almost instantaneously (about 1/20,000 sec.) into an equal volume of gas at a very high temperature and at a pressure of the order of one million lb./in.^{2} The water surrounding the charge possesses both a high density and a low compressibility and confines the gases in a bubble.

If the charge is fired in contact with a ship's hull, then the plating, which can only withstand a relatively low pressure in lb./in.^{2}, is blown in, and the pressure in the explosion is such that, together with the projectile effect of the splinters of plating, further inner plating is pierced unless elaborate systems of protection are fitted. The inrush of gases and splinters followed by water cause extensive local damage to any machinery in its path.

If, on the other hand, the charge is not in contact with a ship, then the bubble of gases expands rapidly outwards, the pressure falling (approximately adiabatically) as the sphere of gases increases in diameter.

The effect of the very rapid initial rise of pressure followed by the expansion of the bubble is to send through the water in all directions a pressure pulse travelling on a spherical front at the speed of sound in water (about 5000 ft./sec.) at a distance from the explosion but considerably higher close to the bubble owing to the high pressures involved.

This pressure pulse has a steep-fronted peak followed by a slower fall as the bubble expands, and as the front of the wave reaches each (spherical) layer of water so it compresses it and gives each particle of water in the layer an outwards velocity proportional to the pressure in the wave at that point. This particle then acts on a particle in the next layer and accelerates it while bringing itself nearly to rest, thus carrying on the wave front. Once the wave front has passed, the water reverts almost to normal, having only a small outward velocity except near the bubble.

Near the bubble, however, the velocity of the water is considerable, and if an explosion occurs close to a ship the pressure in the pulse may well be sufficient to rupture the hull, and the water between the bubble and the hull will be projected inwards, causing considerable damage.

In the general case the bubble continues to expand until the pressure of the gas equals the hydrostatic pressure, but, since at this point the water at the boundary has considerable outward velocity, the bubble carries on over-expanding for some time until the pressure is considerably below hydrostatic. The water then reverses its flow and the bubble contracts, but again overshoots the mark, the pressure rising to a second peak and sending out a secondary pressure pulse.

This oscillation of the gas bubble carries on until, consequent on the upward movement of the bubble due to its buoyancy, it breaks surface. These secondary pressure pulses are readily noticeable by an observer in a vessel close to an under-water explosion like hammer blows on the hull, and three or four are often noticed with fairly deep explosions.

While the pressures associated with the secondary pulses are considerably lower, they may be of importance if the charge is underneath a vessel, while

in certain cases the maximum diameter of the bubble at the end of its initial expansion may be such that the bubble touches the skin of a ship. In this case the ensuing contraction of the bubble will take place against the ship's side, and the maximum pressure in the second pulse may be sufficient to rupture the plating even though the initial pressure pulse was insufficient to do so.

The pressure/time curve of the initial phase has been measured experimentally by several observers, and can be closely approximated to by an instantaneous rise of pressure to the maximum pressure p_{max} followed by an exponential fall giving the pressure at any moment:

$$p_1 = p_{max}\, e^{-nt}$$

where n is a function of the weight and composition of the charge and is given by

$$n = \frac{C}{W^{\frac{1}{3}}}$$

while p_{max} is found to decrease approximately with the distance expressed in charge radii.

Hence $$p_{max} = \frac{KW^{\frac{1}{3}}}{D}$$

where W = equivalent weight of T.N.T.;
D = distance from center of explosion;

and from the experimental results the various constants can be evaluated, and, if great accuracy is required, allowances can be made for dissipation losses. The equivalent weights of T.N.T. for the various explosives used for under-water work can also be obtained experimentally; non-detonating explosives, however, give very low values of maximum pressure owing to the time taken in burning.

Effect at Surface of the Sea.

When the spherical front of the pressure pulse reaches the surface it has to adapt itself to the atmospheric pressure that obtains there, and this can only be done by the reflection of the pulse as a negative pressure, i.e. tensile pulse. This reflection of pressure of opposite sign occurs at free surfaces in all sound and stress wave problems.

As a result of this reflected pulse the pressure in the water just below the surface rapidly becomes negative, and when this tension exceeds a certain value (believed to be about 200 to 500 lb./in.^{2}) the water breaks and the energy in the layer of water above carries the broken water upwards as a spray. This effect is greatest directly above the charge, falling off radially outwards due to the greater distance travelled by the pulse and to the decreasing value of the vertical component. As a result a dome of spray rises with a fairly definite radius and surrounded by a black ring. (Here it is believed breaking of the water is still occurring, but the energy is insufficient to throw the broken water clear of the surface, although sufficient to alter its power of reflection.) The maximum radius and height of this spray dome bear a definite relation to the size and depth of the charge, and from high-speed photographs it is possible to check the depth of the explosion from this data.

Finally, the bubble breaks through the surface and projects upwards the "plume" of water and gas; this plume may vary considerably with the state of the bubble at the moment it breaks through, being most spectacular when the bubble is near a maximum pressure condition on breaking through.

With deeper explosions the dome is smaller, until finally no dome is formed, a shimmer with droplets of spray occurring on the surface as the pressure pulses arrive, while instead of a plume the gases come up in a mass of creamy water.

Effect of Sea Surface and Bottom Reflections.

In the case of charges either on the bottom or close to the surface, the effect on a ship will be altered by the arrival of reflected waves at a short interval after the direct pulse. In the case of reflections from the bottom, the effect will be additive, and in the ideal case of a charge resting on a rigid sea-bed the pressure would be doubled; however, as a rule, the increased effect is small, due to the softness of the sea floor.

With a charge near the surface, a reflected tensile pulse will arrive at the ship before the direct pulse has died away and will tend to cut off the pressure in the pulse, though, since water can only withstand a limited tension, the reduction is not so large as would occur if this were not the case. In addition, with shallow explosions the bubble will break surface and vent before it has fully expanded, and the amount of energy imparted to the water will be less.

Initial Effect on a Ship and Subsequent Whipping.

When the main pressure pulse strikes the hull of a ship, the shell plating being relatively free to move is given a very high acceleration and attains a velocity of some tens of feet per second in a thousandth of a second or less. The plating bends between the more rigid supporting frames, but these in turn gain velocity, the acceleration depending on the masses supported. As the pulse reaches the more distant parts so they are in turn accelerated and gain velocity, until after 20 to 30 thousandths of a second the whole ship has been set in motion. As the pressure dies away at any position, so the elasticity of the members will overcome the pressure and the part will tend to oscillate back to its position of rest at its natural frequency, generally with considerable damping.

If the movement of the plating is sufficient to cause plastic strain, the plates will be dished inwards, giving the typical "hungry look"; severe dishing is usually accompanied by distortion of bulkheads, etc.

As the various sections of the ship are set in motion at different periods of time, the ship as a whole is deformed and, in addition to the local oscillations of plates and frames, larger items, such as bulkheads, will tend to vibrate, and, in turn, the whole ship will vibrate as a beam with two or more nodes. This latter is called whipping, and its magnitude varies considerably with the position of the explosion.

This whipping, though involving considerably lower accelerations than the initial shock on the arrival of the pulse, is associated with large displacements up to 1 or 2 ft., and is the phenomenon most noticeable to the human observer. It may also cause the throwing about of loose objects and damage to items on certain types of mountings. In addition, the interval between successive pulses is often of the same order as the period of whipping of the ship, so that with charges nearly under the ship a considerable increase in whipping may result from the secondary pulses.

Waves of Stress and Their Reflections.

In considering further the transmission of the pulse to parts inside the ship it is necessary to consider the theory of propagation of stress. Thus, if an impactive blow be given to one end of a bar of metal, a wave stress is propagated along the bar at a velocity equal to the velocity of sound in the metal, viz. $\sqrt{E/\rho}$, where E = Young's modulus and ρ its density. This wave of stress is similar to the pressure pulse in the water except that in this case the motion is on a plane front instead of a spherical. Again each particle in being pushed by and pushing on its neighbors attains a momentary particle velocity proportional to the intensity of the stress.

When such a stress reaches a free end, then, as in the case of the free surface of the water, a wave must be reflected back along the bar of opposite sign (pull instead of push) to make the stress zero, and such stress will oscillate to and fro until it dies away. If, on the other hand, it strikes a rigidly fixed end, the

reflected wave will be of the same sign and a force equal to twice the blow acts momentarily on the fixed surface. Similarly, with partially fixed ends and changes of section, varying reflected waves will be caused, and these waves can be superimposed to give the total stress at any instant at any position. Such waves occur even with normal rates of loading, but in view of their velocity (16,700 ft./sec. in steel) the variations in strain caused by them are negligible. Even in the case of the shock wave experienced by light machinery items, the time of application of which is some 2 to 3 thousandths of a second, the effect of such wave propagations of stress does not show on records and can normally be neglected.

Forces Acting on Internal Machinery and Fittings.

The pressure pulse having caused the plating and frames to deflect, these items will in turn exert forces on the various seatings, bulkheads, etc., and hence on the machines and fittings mounted on them. Depending on the items involved, the masses will be accelerated up to appreciable velocities, and as the pulse dies away they are brought to rest by their fastenings or internal stresses.

The resulting effects are many and varied. From the initial pressure pulse we get the very considerable fractures of main and auxiliary machinery and stretching of holding-down bolts, while from the later whipping cases have occurred of priming of boilers, throwing around of floorplates, ladders, and other loosely secured items, and the tripping of governors on dynamos, feed pumps, etc., besides the very considerable effects noticed on electrical machinery and equipment, gun mountings, rangefinders, and the small items such as wireless and radar and their aerials and arrays.

Calculation of Effect on an Item.

In order to be able to design machinery to withstand shock, it is obviously necessary to get a fairly close approximation to the motion undergone by the body, so that a knowledge may be obtained of the accelerations to which a body may be subjected and their duration, as, in general, failure of a body is caused by relative motion; thus, except for the most brittle materials, fracture is always preceded by strain which involves movement in the body.

However, as in the case with all calculations involving complicated bodies, such as ships' structures, there are too many factors about which insufficient data are as yet available to permit of a detailed calculation of shock effects. Such factors are:—

(1) The stiffness of the skin plating and its associated framing and the energy absorbed in dishing of the plating and distorting the frames.

(2) The effect on such stiffnesses of added members, supports for machines, bulkheads, etc., and the resultant motion of the platforms and seating of machines, etc.

(3) The increase in yield point and U.T.S. that occurs when metals, particularly low carbon steels, are subjected to rapid loading.

(4) The effect of the reflections of the pressure pulse from the sea bottom and surface and diffraction of the pulse around corners.

(5) The reflection of the pressure pulse from the moving ship's plating and whether under certain circumstances this leads to cavitation in the water close to the plate—a matter the effect of which is most difficult to assess.

Nevertheless, a reasonably close approximation to the velocity that an item would attain, together with an appreciation of the acceleration, can be obtained from consideration of the behavior of an unrestrained flat plate subjected to a pressure pulse.

In Appendix I the maximum velocity of such a plate is derived in terms of the weight of the charge and its position and also the proportion of the energy

of the pulse that is absorbed by the plate in obtaining its maximum velocity, and the value of this ratio is plotted in Figure 18 of this Appendix for various weights of plate and charge.

Though, in fact, the ship's structure consists of relatively thin plates formed into panels by longitudinals and transverse frames, consideration of the various aspects of the problem as given in the Appendix shows that when considering fairly concentrated items of equipment the error is small if we take the item as equivalent to a plate of the total supported weight divided by the area contributing to its motion. This is particularly so since, for the weights per unit area commonly met with, the proportion of energy absorbed varies only slightly from a mean figure of approximately 0.5. Thus, provided a reasonable assumption of the area contributing to the motion can be made, the maximum velocity of an item can be obtained from a formula

$$\tfrac{1}{2}\,M\,V^2_{max} = \text{constant} \times \frac{W \sin \theta}{D^2}$$

$$V_{max} = \frac{K}{\sqrt{M}} \frac{\sqrt{W \sin \theta}}{D}$$

Such an approximation, however, applies only up to the elastic limit of the structure under consideration, and above this the plastic deformation of the hull plating and frames will result in the absorption of energy that would otherwise contribute to the motion. The effect of plastic dishing is again, as yet, incalculable, and must be founded on practical experiment; since the average ship will only stand a very few shocks of this magnitude this is both long and costly.

Sufficient data are, however, available to give an overall picture of the accelerating forces which items may have to withstand when subjected to a shock wave that just ruptures the hull. Note the greater the mass per unit area of hull the lower the acceleration, while position in the ship has a considerable bearing.

	Accelerations	Duration in thousandths of a second
Shell plating	Several thousand g	1 or less
Items mounted directly on frames	100–300 g	2 to 5
Average auxiliary	100–200 g	3 to 10
Section of ship vibrating ..	50–100 g	5 to 15
Ship whipping as a whole ..	2–5 g	Several hundreds

The motion of typical items such as the above can be seen from Figure 1, showing velocity/time records as recorded by the seismographic type of velocity meter on actual trials.

Design of Machinery to Resist Shock.

Before going into the details of design it is necessary to view the picture as a whole. A warship is designed according to its class to stand a certain

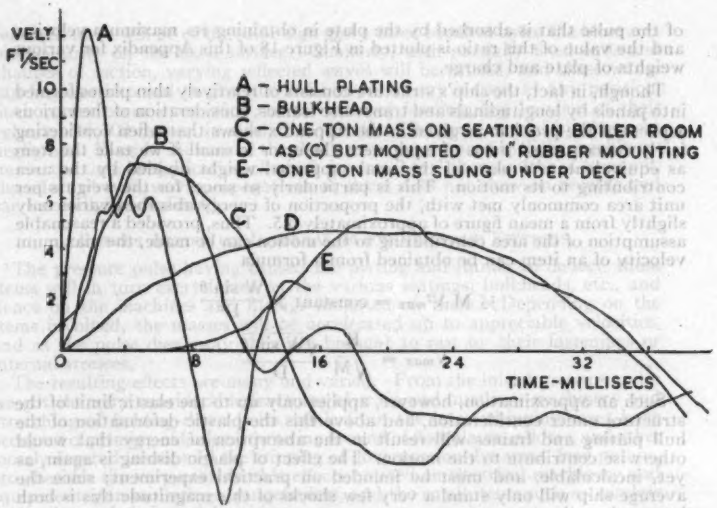

FIG. 1.—TYPICAL VELOCITY TIME CURVES

degree of punishment, and what is desired is that all the functions of a fighting ship can be carried out to the moment when the ship is so badly damaged that she can only limp to a refitting port.

It is essential that no one type of damage shall cripple her at an earlier stage than can be helped, nor should the refitting be unduly prolonged owing to the predominance of damage that cannot be readily repaired. Such damage is illustrated in Figures 2 and 9 of a reciprocating dynamo and main turbine.

It was in this respect that shock damage was so serious, the replacement of large and intricate castings being a lengthy process taking a considerably longer period than the hull repairs. In order to obtain a balance of qualities, it has been decided that items should be able to withstand shocks to the point at which uncontrollable flooding occurs in the compartment in which they are situated.

Since the ship is then in a bad way, no factors of safety are required, and in fact it is probably sound design for certain parts of an item to have yielded at this stage provided the machine will still just function and the yielded parts are easily replaceable.

Design figures for accelerations to be worked to are the next essential, and, as an example, the average auxiliary should be able to withstand

120 g upwards,
60 g downwards,
40–60 g athwartships.

Certain other figures for special cases have been promulgated from time to time, but it is to be hoped that a more comprehensive list of design figures can be promulgated to machinery designers when the experiments now in hand have been completed.

Next, it is desirable to define more closely the above figures of accelerating forces, etc. Thus an auxiliary machine designed to these figures should be capable of withstanding a force equal to, but not exceeding, 120 times its

weight applied to its base for an indefinite period. Next, the base slows down and applies through the holding-down arrangements a retarding force of 60 times the weight of the machine indefinitely. A further requirement is that it can stand the base moving athwartships and applying a force of 40 to 60 times the weight of the machine.

Though, in fact, the accelerations may be of short duration, they may exceed those figures, and they are given as of indefinite period to simplify the design problem.

The stresses caused in the machine by these forces fall into three principal types:—

(1) Direct compressive (or tensile) stress due to the acceleration of a mass through a supporting column; these seldom cause failure.

(2) Bending stresses during acceleration due to the force not acting through the center of gravity of a specific part of the machine—a common cause of failure.

(3) Bending stresses during deceleration due to the holding-down arrangements acting on the feet of the machine and causing these to fracture or the holding-down bolts to stretch—a very common result.

If the materials used in construction are brittle, then fracture will occur, and it was evident from the first that cast iron was the chief sufferer from this type of damage, whereas ductile materials, which not only had a higher strength but could yield and distort slightly without necessarily putting the machine out of action, were seldom affected seriously except in cases of very poor design.

The prohibition of cast iron for all except a few items where its properties merit its special retention under due safeguards was the first obvious step, but could not be fully implemented owing to production difficulties in wartime. Cast aluminum is a similar source of weakness; gunmetal is better, provided a reasonable ductility is assured.

In addition, then, to the avoidance of brittle materials, shock design requires:—

(1) The placing of the feet or supporting chocks and the design of the base so that the forces are transmitted directly to the main mass of the machine.

(2) Design of the feet or bedplate and its holding-down arrangements so that the decelerating forces will not fracture or distort the main body of the machine. Stretching of holding-down bolts or distortion of keeps is probably acceptable.

(3) Suitable design of all overhanging masses and their supports to avoid excessive bending moments.

(4) Dynamic balance of trip gears and similar mechanisms so that shock does not tend to trip turbo-generators, etc.

Design for (1) and (3) is not as a rule difficult, while (2) can be safeguarded against by making the holding-down bolts with adequate stretching lengths and considerably weaker than the foot; similarly, keeps can be designed so that they will yield before the foot breaks (see Appendix II). Thus, if 16 tons/in.^{2} be allowed for a mild steel foot that should not yield, then the ho'ding-down bolts or keeps should be stressed to 25 tons/in.^{2} under these conditions to ensure yielding. Even higher stresses would not be amiss, for, owing to the short duration of the decelerating period, the actual elongation will be very small.

It is also good practice to mount auxiliary machinery on a fabricated bedplate or deep girder so that any slight movement of the base does not cause misalignment.

Main Turbines.

As can be seen by inspection of a turbine set and its mountings, the normal turbine is ill-suited to withstand shock forces, and with the cast-iron construc-

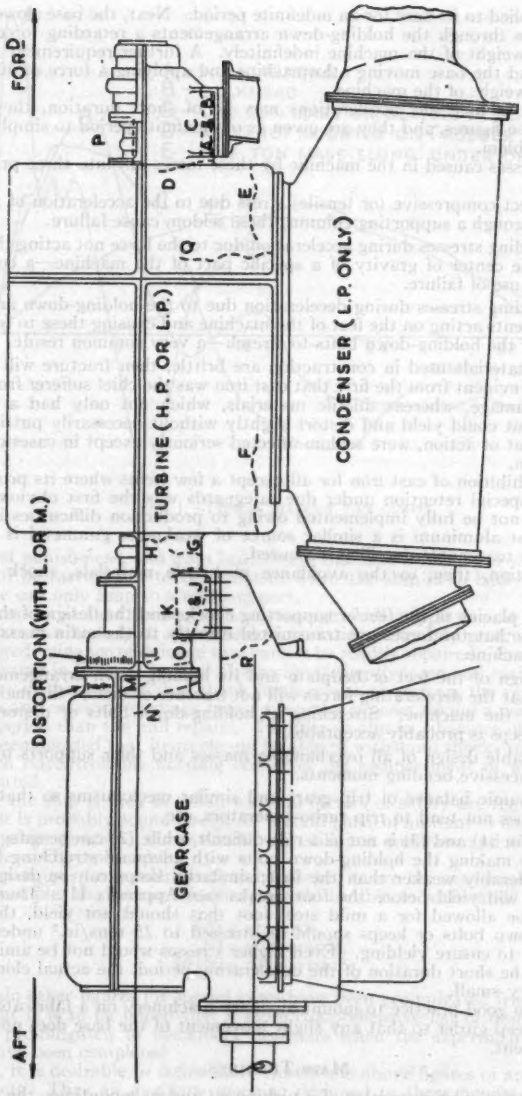

FIG. 12.—DIAGRAMMATIC ILLUSTRATION OF MAIN TURBINE SET, SHOWING FRACTURE

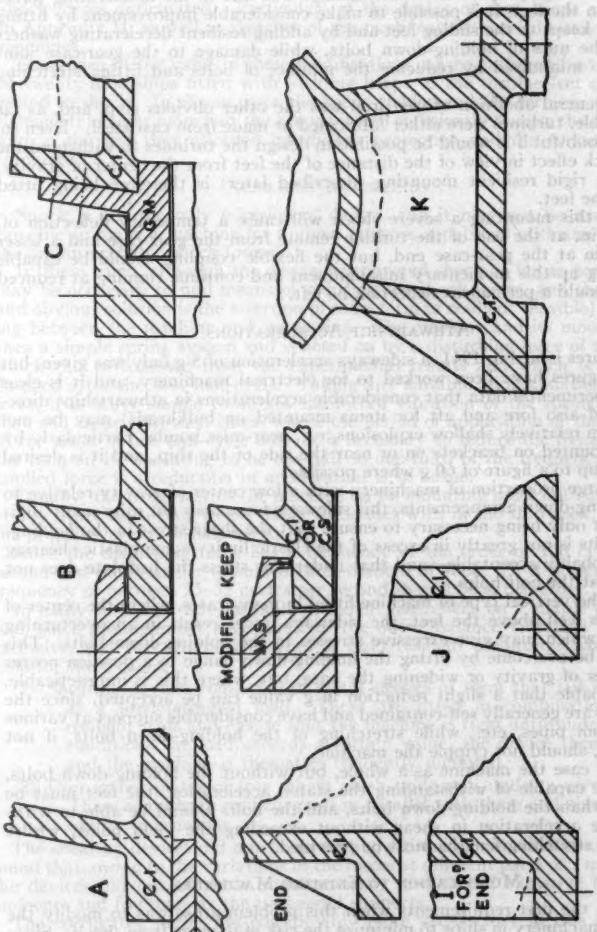

FIG. 13.—ILLUSTRATION OF FRACTURES ON FIG. 12

C.I. Cast iron. C.S. Cast steel. G.M. Gunmetal. M.S. Mild steel.

tion in general use at the beginning of the war many turbine units were wrecked by the magnetic mine and near-miss bombs.

Since a large proportion of the damage was due to the decelerating forces acting on the feet, it is possible to make considerable improvement by fitting yielding keeps to the sliding feet and by adding resilient decelerating washers under the nuts of holding-down bolts, while damage to the gear-case joint was also minimized by reducing the number of bolts and fitting stretching lengths.

The general abolition of cast iron was the other obvious step, and, as far as possible, turbines were either fabricated or made from cast steel. Even so, it was doubtful if it would be possible to design the turbines to withstand the full shock effect in view of the distance of the feet from the center of gravity, and the rigid resilient mounting (described later) is therefore being fitted under the feet.

With this mounting a severe shock will cause a temporary deflection of, say, ¼ in. at the end of the turbine remote from the gear case and a lesser deflection at the gear-case end, but the flexible coupling should be capable of taking up this momentary misalignment and continue running at reduced speed should a permanent deflection be left.

Athwartship Accelerations.

In figures issued in 1941, a sideways acceleration of 5 g only was given, but higher figures have been worked to for electrical machinery, and it is clear from experimental data that considerable accelerations in athwartships direction (and also fore and aft for items mounted on bulkheads) may be met with from relatively shallow explosions, e.g. near-miss bombs, particularly by items mounted on brackets on or near the side of the ship, and it is desired to work up to a figure of 60 g where possible.

In a large proportion of machinery with a low center of gravity relative to the holding-down arrangements, this sideways force does not raise many difficulties, it only being necessary to ensure that the shear stress in the holding-down bolts is not greatly in excess of the elastic limit—some plastic shearing being probably acceptable—and that under this stress the bedplate does not fracture at the bolt holes.

With the vertical type of machine fitted in many cases, where the center of gravity is well above the feet, the sideways forces result in an overturning moment which may give excessive stresses in the holding-down bolts. This can best be overcome by lifting the holding-down flange to a position nearer the center of gravity or widening the base, but, where this is impracticable, it is probable that a slight reduction in g value can be accepted, since the machines are generally self-contained and have considerable support at various levels from pipes, etc., while stretching of the holding-down bolts, if not excessive, should not cripple the machine.

In this case the machine as a whole, but without the holding-down bolts, should be capable of withstanding the stated acceleration, the feet must be stronger than the holding-down bolts, and the bolts should be able to withstand the acceleration in shear without exceeding the yield point, whilst adequate stretching lengths must be provided.

Modification to Existing Machinery.

One of the first requirements when this problem arose was to modify the existing machinery in ships to minimize the risk of damage from shock. Since major alterations could not be made, all that was generally possible were makeshift arrangements.

These consisted of the alteration of existing chocks and the addition of extra chocks—in many cases of wood—under the main mass, or the webs supporting it, so that the forces were transmitted more directly. Overhanging masses had separate supports arranged or were bracketed on to the main body (even this was not always sufficient), while the feet were protected by the pro-

vision of stretching lengths on the holding-down bolts, placing flexible washers under the nuts and alterations to keeps so that they would yield under the shock forces anticipated. Particulars of the design of such keeps are given in Appendix II, as well as sketches of the modifications carried out to typical machines.

Some idea of the value of these modifications can be gathered from reports of twenty-nine ships fitted with yielding keeps to the turbine feet and subjected to shock. In eleven cases the keeps yielded and the turbines were not damaged; in eight more half the keeps yielded sufficiently to save the turbines, and in the rest they afforded insufficient protection and the turbines were fractured.

Resilient Mountings.

Shock is not the only criterion of design, and it may not be reasonably possible to make a machine for a given purpose capable of withstanding the maximum probable shock forces.

In these cases, although the design has covered shock as far as possible, it may be necessary to find means of reducing the shock forces, and the simple and obvious solution is the insertion of some form of resilient (flexible) mounting between the machine and its seating. The machine and its mounting is then a simple spring system and is acted on by a disturbing force of a period which can be assessed from the experimental results and which is heavily damped after the first half-cycle. Taking any suitable approximation to the force, it is fairly simple to obtain the forced vibration of the spring system that will ensue. In such cases, where the period of application of the load is of the order of one half-cycle only, it is necessary to reduce the frequency of the item on its mounting to be one-half to one-third of the frequency of the applied force if a reduction in acceleration is to ensue.

Equally, the frequency of the machine on its mounting must not equal that of any other disturbing forces likely to be met with in the ship itself, such as the propeller or out-of-balance forces.

Since the disturbing force has a frequency of the order of 100 cycles per second and the propeller (assumed three-bladed) of about 15, it is seen that a frequency of between 25–35 cycles per second should be satisfactory.

In many cases it is not desirable to mount machinery on a simple resilient pad, since misalignment may result or the out-of-balance vibration may be excessive, and two alternative types of rigid/resilient mountings have been used for these cases.

(1) Shearing devices consisting of metal sleeves of form that fail in shear at a predetermined load. These are fitted in combination with resilient pads and ensure that the machine is rigid until under shock conditions the load exceeds a certain value; the device then shears and the machine is thereafter resiliently mounted.

(2) Corrugated plates of iron which crush uniformly under a given load. These again have resilient pads in parallel with them and also absorb energy in crushing.

The shearing devices did not prove highly satisfactory in service, as it was found that, owing to the variations in the forces at different parts of a machine, the devices did not all shear simultaneously and caused excessive bending moments and fractures at the unsheared supports.

The corrugated plates are considerably more satisfactory in this respect, since the variations in deflection of different parts of a given machine are smaller, but, nevertheless, relative deflection will occur both between the machine and its seating and possibly between parts of the machine itself, and consideration must be given to this before fitting such mountings.

With all these mountings the resilient pads are designed to bring the machine back approximately to its original position except for any permanent set due to excessive loading. Further, the resilient pads themselves may be liable to creep under permanent loading and cause misalignment.

The fitting of rigid/resilient mountings needs care if good alignment is to be maintained, and a high standard of accuracy in the plates and seatings is required.

Choice of Materials for Resilient Pads.

For this particular purpose a spring is required that will deflect, say, ⅛ in. to ½ in. under a load of, say, forty times the weight of the machine. The material must have a sufficiently long life and resistance to abrasion, oil and heat, and not be subject to creep under prolonged loads. A fairly high damping factor is also useful in preventing resonance conditions and also in absorbing part of the energy transmitted.

Since weight is an all-important factor, a simple rubber pad loaded in compression is in general the most suitable, the strain energy stored per unit volume being greater than in a spring involving bending stress. Steel springs are not satisfactory for this reason and are cumbersome. The use of a plain compression mounting also avoids any bottoming effects with a sudden sharp rise on the accelerations.

Many types involving rubber in shear have also been suggested, but this involves considerable reliance in the bonding of metal to rubber, which, in view of shipboard conditions, is undesirable unless any appreciable advantages are gained.

For heavy items a fairly hard rubber is desirable to keep areas reasonable, and types such as Dexine 118 or Walker's Questo, having a shore hardness of 80–85 deg., are suitable. With these, the compression curves are fairly straight up to 30 to 40 per cent compression, when they gradually steepen. If noise suppression is also important, or when dealing with light and fragile parts, a much softer rubber is needed.

Nevertheless, deterioration of the rubber is a problem, and, though coated with oil-resisting varnish and fitted in oil shields, their life under adverse conditions is not good. The exercise of a little care in design would, however, obviate much of this—one cannot expect a rubber pad to last when placed inside the saveall of a lubricating or fuel oil pump, particularly when fitted with oil shields, which in this case act as oil baths for the rubber.

Rubber Resilient Pads.

In theory, all that is required is to obtain a mounting of a known frequency when loaded with a known mass, but the use of material such as rubber leads to further complications in that the load deflection curve is not straight, while other factors have a considerable influence on the deflection under load, namely:—

(1) Form factor—a plate of rubber with a large area under load and small free surface will be in effect stiffer than in a case where by cutting the plate into pieces the free surface is increased.

(2) Effect of friction between surfaces of rubber and feet of item. The stiffness of a rubber mounting varies considerably between the extremes of well vaselined smooth surfaces and rough machined ones. A bonded rubber-to-metal joint would have the advantage of constancy but would lose the damping effect due to this friction.

(3) Effect of dynamic loading on the frequency of rubber mountings. It is found that a rubber mounting when acted on by a pulsating load has a frequency that increases with the frequency of the load.

As a result of these conflicting factors, it is only possible to specify an approximate area/ton supported for these mountings, based on experimental results, and further investigations are in hand to obtain check figures.

Decelerating Washers and Stretching Lengths in Holding-Down Bolts.

The placing of a simple rubber pad under the holding-down bolts (between suitable metal washers) was an obvious means of lessening the decelerating

forces on feet and holding-down arrangements. This in many cases was all that was necessary to safeguard the machine since the feet in decelerating are usually the weakest part. The areas of rubber fitted are generally lower than that of the accelerating type of pad, as the maximum decelerating forces met with are usually lower than those in acceleration.

Since no weight is taken on the washers, questions of creep and vibration do not normally arise, but the washers should not be screwed down unduly tightly when fitted.

Another method of limiting the decelerating force on the feet is the use of stretching lengths in the holding-down bolts, and this can be fitted in conjunction with decelerating washers or without them where a more rigid arrangement is required.

The diameter of the stretching length should be sufficiently below the diameter at the bottom of the thread to ensure that notch effect does not cause fractures in the thread. The elongation of the stretching length will absorb a very considerable amount of energy before fracture occurs, besides making it simpler to design the feet so that fracture of the feet shall not occur. In order to ensure this, it is usual to allow a figure of 24–25 tons/in.^{2} for the yield point of mild steel for bolts that are to yield, while allowing 16 tons/in.^{2} for the feet that should not yield.

Dynamic Yield Strength of Materials.

It has for some time been known that the rate of application of the load affects the yield point of a metal quite considerably. Since this is an important factor in cases such as those under discussion, investigations have been carried out to find the yield point at various rates of loading, and the principal results to date have been published in various technical papers. Further tests are envisaged in which not only the rate of application, but the duration of the load, is varied.

For the moment, however, it seems best to leave this increase of permissible stress as a margin of safety in the design.

Shock Testing Machinery.

In view of the complex nature of the problem, it is clear that calculation alone cannot ensure a wholly satisfactory design, and some means of testing typical machines up to the design limit is necessary. Full-scale tests on ships are not satisfactory for this, since the vessels chosen, and their machinery, are normally obsolete, nor do more than one or two items get the full intensity of shock from a damaging shot.

The ship trials do, however, give us the necessary data on shock effects if we can find some means of reproducing these in a test shop.

Shock machines for testing electrical gear up to ½ ton in weight have been installed at the Admiralty Experimental Laboratory. These are of the swinging hammer type, the intensity of shock being governed by the height the hammer drops.

This gives satisfactory results for the smaller items of gear and has enabled considerable investigation to be made into the movements of switchgear under shock, and very considerable improvements in design have resulted from its use.

For the heavier machinery items, however, the problem is somewhat more difficult, but endeavors are being made to design a testing machine suitable for this purpose on which typical machines could be tested out and the most economical shock resistant design worked out.

Conclusion.

Though there is much yet to learn concerning the forces brought into play by non-contact explosion, it is possible to make reasonable approximations to their magnitude and duration. The design of machinery to withstand such forces then requires only the application of straightforward principles of mechanics to obviate the major casualties that have occurred due to this cause.

The design problem is greatly simplified if brittle materials can be replaced by ductile ones, but where other considerations render this course undesirable much can still be done by avoiding the obvious causes of fracture, some of which are illustrated in Appendix II.

Resilient mountings are a further step in the safeguarding of machinery, but in their turn involve upkeep problems, and the possibilities of improved design to lessen this must be borne in mind. Progress, owing to the difficult nature of trials, must, however, be slow, and it is essential to do what is possible now to improve design against this danger if serious damage in any future conflict is to be avoided.

Appendix I.

Calculation of the Effect of a Pressure Pulse on an Unrestrained Flat Plate and Its Relation to the Effect on Items of Machinery and Equipment.

The motion of a machinery mass in a ship acted on by a pressure pulse will involve the masses of the plating, framing, seating, and the machine itself, together with the stiffnesses of:—

(1) The seating relative to the framing,

(2) The framing to which the seating is secured relative to the hull as a whole,

(3) The panels of plating,

and will thus form a complex spring system the mathematical solution of which does not enable us readily to interpret the motion of the mass.

Partial solutions in which some of the components are considered negligible provide some indication of the motion, but are still complex. It would, however, appear that a useful insight into the motion can be obtained from consideration of the simple case of the pressure pulse acting on an unrestrained free plate followed by an appreciation of the effect on this motion of the various other factors involved.

Pressure in Water.—Neglecting the enhanced values of velocity of pulse, etc., that occur close to the charge, there is formed from an under-water explosion a spherical pressure pulse radiating outwards at the speed of sound in water (5000 ft./sec.) and having a pressure at any instant:—

$$p_1 = p_{max} e^{-nt} \text{ where } p_{max} = C_1 \frac{W^{\frac{1}{3}}}{D} \text{ tons/in.}^2$$

and $$n = \frac{C_2}{W^{\frac{1}{3}}} \frac{1}{\text{sec.}} \text{ and } W = \text{weight of explosive.}$$

The values of C_1 and C_2 being obtained experimentally.

Action of Pressure Pulse on Free Plate.—This pressure acting on a free plate, mass m per unit area, is in part reflected, and the combined force due to the incident and reflected pressures accelerates the plate; thus

$$p_1 + p_2 = m \ddot{x}$$

while from consideration of the particle velocity of the water being equal to the plate velocity $p_1 - p_2 = \rho c \dot{x}$, ρ and c being density and velocity of sound for sea water.

Hence $$m \ddot{x} + \rho c x = 2 p_1 = 2 p_{max} e^{-nt}$$

Resultant Motion of Plate.—Solving the above equation of motion gives

$$\dot{x} = \frac{2 p_{max}}{\rho c - m n} (e^{-nt} - e^{-\rho ct/m})$$

and the pressure and velocity will vary with time, as shown in Figure 17. The velocity is a maximum at

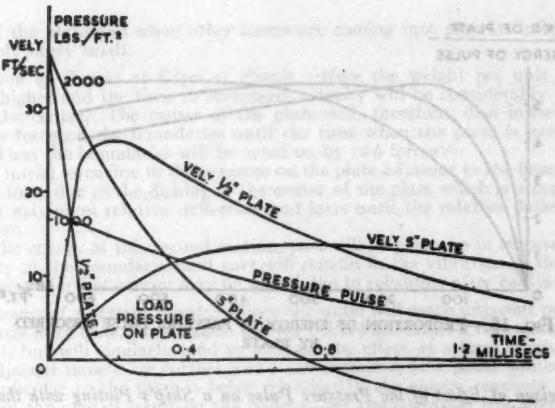

FIG. 17.—MOTION OF PLATE ACTED UPON BY A PRESSURE PULSE FROM A 250 LB. CHARGE OF TNT 80 FT. AWAY

$$t = \frac{1}{n} \cdot \frac{\log_e Z}{Z - 1} \text{ where } Z = \rho c / m n$$

The pressure on the plate is then zero, and the value of the maximum velocity is given by

$$\frac{2 p_{max}}{\rho c} \cdot Z \frac{1}{Z - 1}$$

Proportion of Energy of Pressure Pulse Absorbed by the Plate.

The energy in the pressure pulse in the water is given by

$$\int_0^\infty \frac{p^2}{\rho c} dt, \text{ where } p = p_{max} e^{-nt}$$

$$= \frac{1}{2n} \frac{p^2_{max}}{\rho c}$$

while the K.E. of the plate at maximum velocity $= \frac{1}{2} m V^2_{max}$

Hence proportion of energy absorbed by plate

$$= m \cdot \frac{4 p^2_{max}}{\rho^2 c^2} \cdot Z^{\frac{2}{1 - Z}} \cdot \frac{n \rho c}{p^2_{max}}$$

$$= \frac{4 m n}{\rho c} \cdot Z^{\frac{2}{1 - Z}}$$

$$= 4 Z^{\frac{1 + Z}{1 - Z}}$$

Thus the proportion of energy absorbed by a free plate is a function of the thickness of plate and charge weight, and Figure 18 gives values of this ratio for typical charge weights.

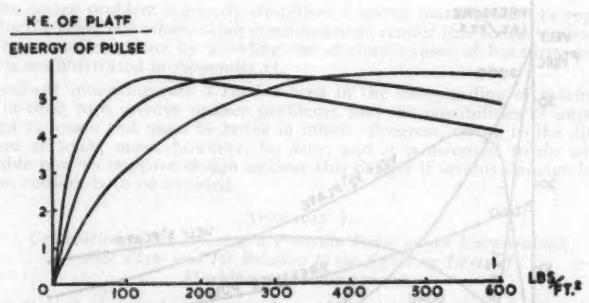

FIG. 18.—PROPORTION OF ENERGY IN PRESSURE PULSE ABSORBED BY PLATE

Comparison of Effect of the Pressure Pulse on a Ship's Plating with that on a Free Plate as above.

The various experiments carried out on ships show that over the range of mass loading and charge weights employed there is a definite relationship between the velocity of items and the energy in the pressure pulse, the energy transmitted being of the order of ⅓. This agrees closely with the proportion of energy absorbed for similar cases given in Figure 18, so that it seems desirable to investigate further the complex motion of a ship's plating to see if some such simple generalization does sufficiently cover the actual state of affairs to permit its use within limits.

Motion of Ship's Plating and Structure.—The hull plating of a ship may have considerable curvature, particularly in the transverse plane, and is divided into panels by transverse and longitudinal frames. These frames will in turn be the foundation of seatings for masses or bulkheads. Taking a specific seating for a mass secured, say, to two consecutive frames, then the adjoining half panels will contribute to the motion. The area contributing to the motion is generally easier to assess as regards the transverse frames than the longitudinals, where the curvature of the plate makes a fair estimate of areas involved more difficult.

For a panel of average size and an explosion at a non-lethal distance and on a line normal to the plate, the pulse will reach all points on the panel simultaneously, and each part will start as a free plate until there is sufficient relative deflection to cause other forces to act.

Motion of Centers of Panels.—The plate at the center is of small mass and relatively lightly constrained at the start, and will rapidly reach its maximum velocity with only a small displacement relative to its edges. Once it has reached maximum velocity the pressure in the water tends to become negative and cavitation may set in, the water breaking at a point close to the plate. The plate, if undisturbed, would then dish inwards to a maximum relative deflection at a quarter of the period of oscillation of a plate constrained at the edges (this is not strictly true, since the time to maximum velocity may not be very small compared to the quarter period). However as the plate comes to rest in space (which occurs at a point after maximum deflection), any cavitation will close up, if this has not occurred previously due to the arrival of diffracted pressure waves, and the quarter cycle back to zero relative deflection will be prolonged and the vibration after the point of maximum deflection heavily damped by this bringing to rest of the cavitated water. This reloading after cavitation, in theory, increases the energy absorbed by the plate, but since it occurs after the maximum relative deflec-

tion of the plate and when other forces are coming into play its influence is probably fairly small.

Motion of Frames at Edges of Panels.—Here the weight per unit area is much higher and the time to maximum velocity will be considerably greater than the center. The center of the plate will, therefore, dish inwards and exert a force on the boundaries until the time when the plate is once more flat. Thus the boundaries will be acted on by two forces:—

The initial force due to the pressure on the plate adjacent to the boundaries.

The force due to the dishing of the center of the plate which is a maximum at maximum relative deflection and lasts until the relative deflection is zero.

Of the energy of this second motion, part will be given up in increasing the velocity of the boundaries and part will remain in the vibration of the plate, while some further energy may be added due to reloading after cavitation.

Influence of Adjacent Panels.—Since the pulse will reach adjacent panels at a slightly later time, these panels will lag behind and exert a restraining effect at first, but will similarly tend to increase the effect at a later stage. Since the adjacent panels are further away (an explosion at a point subtending a perpendicular to the plate is being considered), the energy reaching them is less and the overall result on the panel under consideration is a slight delay in reaching the peak velocity together with a slight loss of energy to adjacent sections.

Summary.

It is seen that the time to maximum velocity of an item in a ship is greater than that of a uniform flat plate owing to the effect of the dishing of the panels, and this is further prolonged by the influence of adjacent panels and possibly by reloading after cavitation.

However, the energy absorbed by the mass, its seating, and the plates and frames does not differ very much from the case of the uniform plate, the variations being:—

Less energy is absorbed by the light central portion of a panel.

More energy is absorbed by the heavy edges of a panel.

Energy is left in the panel vibrating between the frames.

Further energy is added by reloading after cavitation.

Assuming these effects approximately balance, the energy of the item and its associated framing and plating may be equated to a proportion of the energy impinging on the area contributing to the motion as given in Figure 18 and approximating to one-half over a wide range. This is found to be in agreement with trial results.

Although the similarity to the free plate only holds for a short period of time, it can also be used to illustrate another aspect of the motion involving energy principles, namely:—

Oblique Incidence of Pressure Pulse.—In general the line from the charge to any item will not be normal to the plating under the items, but, from hydrostatic principles, it is clear that the pressure in the water must act normally to any surface with which it is in contact.

Secondly, the energy radiated in a pulse is uniformly distributed over a spherical layer so that, whatever the angle between the plating and the line to the charge, only a certain amount of energy proportional to the area normal to the line to the charge falls on the plate and gives rise to velocity normal to the plate.

When, therefore, considering the motion of an item on a flat plate, its velocity normal to the plate should vary as $\sqrt{\sin\theta}$, where θ is the angle between the plate and the line to the charge. In the other extreme case of a circular hull moving as a solid body, the motion normal to the hull of an item will vary as $\sin\theta$, where θ is the angle between line from item to center and line at center normal to direction of charge.

In either case the pressure pulse diffracts round corners and various reflections will be obtained from earlier impingements interfering with the pulse, particularly if cavitation occurs, so that the above figures for the effect of angle will not be so accurate, particularly where θ is small.

Accelerations.—In addition to the peak velocity of an item, an approximation to its acceleration up to this point is required. Here the free plate does not act as a useful guide, and it is only possible to assess this acceleration as the result of the two forces previously noted.

(1) Force due to pressure pulse on the plate adjacent to the boundaries. This will cause an acceleration of $2p_{max}/m$ at the start, falling to zero before maximum velocity, but since no relative motion has occurred the area concerned will be small and m high.

(2) The force due to dishing, starting from zero and working up to a peak and then back to zero. The acceleration due to this will, therefore, be a function of the period of oscillation of the plate.

These forces, one diminishing and the other increasing to a peak and then falling, followed by the influence of adjacent panels, give rise to a fairly constant acceleration over the initial stage followed by a diminishing acceleration up to maximum velocity, and this is typical of most velocity meter records obtained.

The initial value of the acceleration due to (1) is not readily assessed, but will obviously be greater for strong framing, e.g. submarines, than for weak. For other than such specific cases it would appear that to a first approximation the accelerations are comparable on a basis of the period of oscillation of the plating and mass loading.

Appendix II.

Modifications to Existing Machinery.

As soon as the possibilities of damage due to shock had been appreciated, steps were taken to put in hand modifications to minimize the likelihood of damage to existing machinery.

Since one of the major causes of damage was the decelerating forces causing fracture of the feet, action was taken to modify the keeps and holding-down arrangements so that these parts should be weaker than the feet.

Damage was particularly frequent at the sliding feet end of main turbines, while fracture of these led in many cases to damage at the connection to the gear case, and the sliding feet keeps were therefore modified on the following lines.

Design of Yielding Keeps.

So as to avoid an excessive bending movement in the foot, the protruding part of the keep had to be designed to bend at a safe load. Equally, the deflections had to be kept to a minimum and the greatest amount of energy was required to be absorbed in the bending of the keep. This condition was satisfied by making the keeps of mild steel and designing the protruding part to a parabolic contour or a close approximation to it.

In addition, the holding-down bolts of the keep should not yield before the keep itself, and to satisfy these requirements the keeps were made as shown in Figures 19 and 20, the dimension a being given by the two formulae for which equal values of a should be obtained.

$$a = 0{\cdot}54t \sec\theta \sqrt{\frac{b}{c + t/3 \tan\theta}} \quad \text{(bending of keep)}$$

(θ as in Figure 20)

$$a = 0{\cdot}5 \sqrt{\frac{A\,e\,b\,N\,p}{l\,f}} \quad \text{(stretching of bolts)}$$

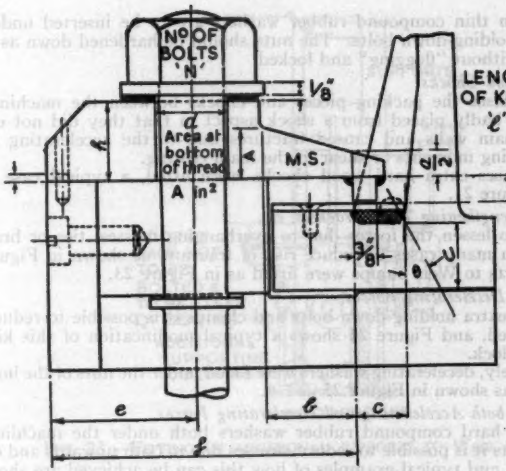

FIG. 19.—DESIGN OF YIELDING KEEP

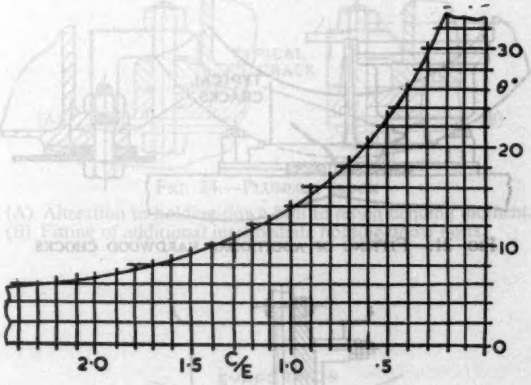

FIG. 20

The dimension b should if possible be not less than a, and the product $a\ b$ as large as possible and in general not less than 0·75 in.²

Value of p in above formula should be taken as 16 tons/in.² for mild steel bolts up to 40 tons/in.² for H.T. steel bolts of U.T.S. 55/60 tons/in.²

The thickness of h of the keep should be made

$$= 1{\cdot}1\ a \sqrt{\frac{l\,(f-a)}{b\,(l - \mathrm{N}\,d)}}$$

In addition thin compound rubber washers are to be inserted under the nuts of the holding-down bolts. The nuts should be hardened down as much as possible without "flogging" and locked.

Fitting of Wood Chocks.

In many cases the packing pieces and chocks between the machine and seating were badly placed from a shock aspect in that they did not extend under the main webs and caused fractures due to the accelerating forces causing bending moments or shear in the main casting.

In such cases extra hard wood chocks were fitted, a typical case being shown in Figure 21.

Fitting of Strengthening Ties, Brackets, etc.

In order to lessen the forces due to overhanging masses, ties or brackets were fitted in many cases to reduce risk of fracture, as shown in Figure 22. Extra supports to Weir pumps were fitted as in Figure 23.

Reduction in Decelerating Forces.

By fitting extra holding-down bolts and clamps it is possible to reduce the forces involved, and Figure 24 shows a typical modification of this kind to a plummer block.

Alternatively, decelerating washers were fitted under the nuts of the holding-down bolts, as shown in Figure 25.

Reduction in both Accelerating and Decelerating Forces.

By fitting hard compound rubber washers both under the machine and under the nuts it is possible to reduce stresses due to both upwards and downwards forces, and typical examples of how this can be achieved are shown in Figures 26 and 27.

Where the presence of unbalanced forces makes this type of mounting unsuitable, rigid resilient mountings are fitted instead.

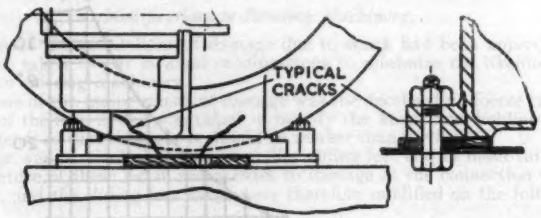

FIG. 21.—FITTING OF ADDITIONAL HARDWOOD CHOCKS

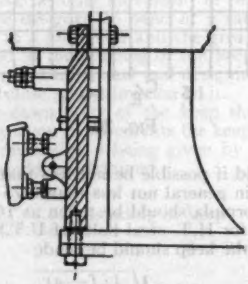

FIG. 22.—FITTING OF STRENGTHENING TIES TO TURBINE YOKE

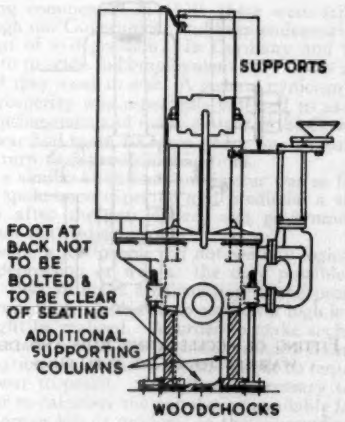

FIG. 23.—FITTING OF ADDITIONAL SUPPORTS TO WEIR PUMP

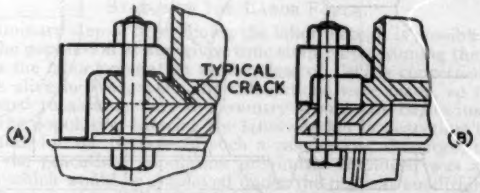

FIG. 24.—PLUMMER BLOCK

(A) Alteration to holding-down bolt to lessen bending moment.
(B) Fitting of additional intermediate holding-down bolts.

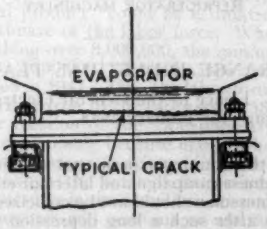

FIG 25.—FITTING OF DECELERATING WASHERS UNDER NUTS

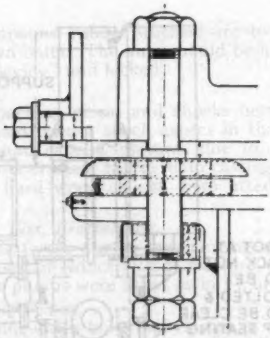

FIG 26.—FITTING OF ACCELERATING PADS AND DECELERATING WASHERS TO AN OIL FUEL PUMP

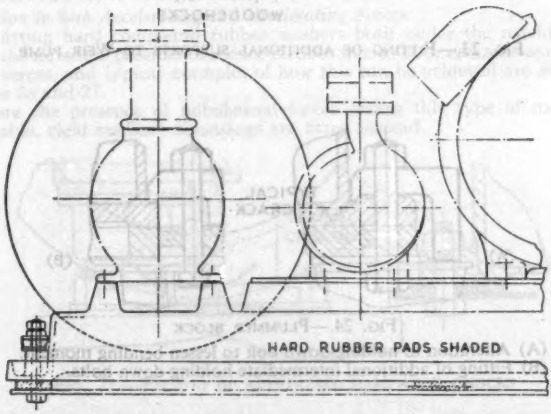

FIG. 27.—FITTING OF ACCELERATION AND DECELERATION WASHERS. REFRIGERATOR MACHINERY

LONG-RANGE INDUSTRIAL PLANNING.

This artic'e by D. C. Frince of the General Electric Co. is reprinted from "Mechanical Engineering" for January 1948.

All during the 1930's this country experienced varying degrees of depression. Then, with our preparedness campaign and later our entry into the war, things began to pick up. Businessmen should not be criticized too severely if to them this pickup of business after such a long depression was construed to mean that, at the end of the war, we might look forward to another depression with further unemployment, and that such conditions might only be alleviated for any considerable time by some other war or threat of war.

When this upswing commenced in 1940, there were still over 8,000,000 unemployed—although our Government had been endeavoring for some eight years to bring us out of a depression. In Germany and Italy, Hitler and Mussolini had tried to provide full employment by various national-planning schemes. In the end they went to war. A general cynicism covered our own land. The rise of prosperity was repeatedly referred to as a "phoney prosperity," and many businessmen and many people in the Government believed that as soon as the war had spent its force, there was almost no likelihood of anything except a return to depression conditions.

A conviction that a similar situation would recur was so firmly rooted that various government spokesmen expected and predicted a similar amount of unemployment even after the war ended, and government policies were definitely predicted on that expectation.

In spite of these fears, a few people did not see any logical reason why we need regard either depression or war as the only possible alternative. It appeared possible in 1941 to make a calculation of the productive potential of the country and from that calculation to see how a high level of production and employment might be realized. In order to make such a calculation, it was assumed that the war would run an indeterminate number of years and that upon the termination of the war two years might be required to reconvert our industries from war to peace. Since it was necessary to have an actual year in mind, in order to calculate the population available for work, the year 1947 was selected, more or less at random, on the assumption that war would end in 1945. Actually, of course, the war did end in 1945, so that the two years have now elapsed. Our reconversions are practically complete, and therefore it should be possible to analyze the accuracy with which such analyses were made at that time.

Status of the Labor Force.

As a preliminary step in determining the labor force, it is possible, of course, to predict the population at any given time simply by assuming the birth rate. In so far as the labor population at the close of 1947 is concerned, all those people were alive in 1941, when the calculations were made, so that it was only necessary to assume that the country would not be decimated. The fraction of the population entering the labor market in peacetime is a matter of record; and, for the purpose of such a calculation, therefore, the highest fraction of the peacetime population previously employed was assumed as the fraction which would be employed under the postwar conditions.

The productivity per individual worker was assumed to be the same as immediately before the war, the theory being that, during the war, although productivity of war items might greatly increase, still the reconversion period back to things of peace which had not been made during the war would find the efficiency of production on approximately the same level as it had been immediately before the war effort was undertaken. Multiplying the number of the probable labor force by the probable productivity per individual worker, a "gross national product" could be estimated.

Figure 1 shows the estimate of the labor force. Whereas the unemployed in 1940, numbered something over 8,000,000, the minimum visualized for the year, $V + 2$, was set at 2,500,000. Actually, the record shows that only 2,300,000 were unemployed in 1947. Also, the labor force actually approached the 60,000,000 mark, as compared to the estimate of 58,000,000. This represents a somewhat higher fraction of the total population in the labor force than had been expected, presumably because many, who had started working during the war for patriotic reasons, continued in order that they might have the added earnings. Although the total of the estimate is fairly accurate, there are considerable departures in the distribution, and the question naturally arises whether these departures will continue or whether a further readjustment will take place back toward a more normal distribution of labor.

The most unexpected variation is that the labor force on the farm has shrunk some 28 per cent from the predicted value. It is actually less than it was in

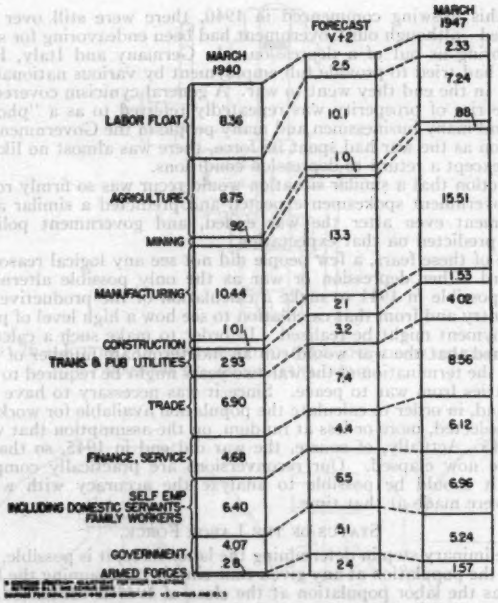

FIG 1 COMPARISON OF EMPLOYMENT FORECAST WITH ACTUAL 1947.
IN MILLIONS

1940. Apparently, a great many of those who went into war industry have preferred to stay in manufacture; and it will be noted that the corresponding figure for those engaged in manufacture is considerably higher than the expected figure. This is probably to be accounted for in some measure by the fact that, immediately after the war, we had very large deferred demands, and it has been necessary, therefore, to manufacture not only for the current year but to make good the deficits of the war years.

The Gross National Product of Labor.

The gross national product resulting from the work of these people appears in Figure 2. The first column in this chart shows the actual performance in 1941. The second column is the forecast for the year, $V + 2$. Acting on the theory that there would be a large amount of unemployment immediately after the war, unless purchasing power were maintained, labor was encouraged to ask for very considerable wage increases. It should have been obvious that there was no immediate need for these wage increases inasmuch as all during the war a great deal more money had been handed out in wages than the people had been permitted to spend on account of the shortage of materials of all sorts. Therefore, these savings hung over the market and, when the wage increases were added to them and the price ceilings were removed, it was inevitable that prices should increase considerably.

Since our estimate of gross national product is based on the man-hours and production per man-hour, it was therefore necessary to make a correction to

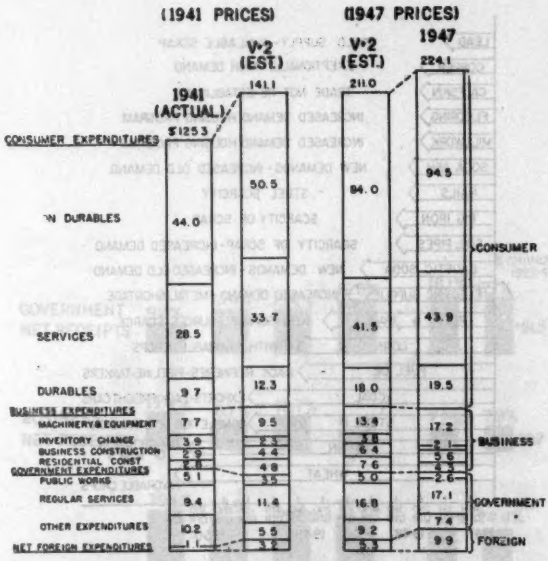

FIG. 2 COMPARISON OF GROSS NATIONAL PRODUCT FORECAST WITH ACTUAL 1947, BILLIONS OF DOLLARS

the "gross national product" resulting from these various price and wage increases. This has been done in column 3. This projection gives a total estimated "gross national product" of some 211 billion dollars. Actually, it appears that the realized figure for 1947 will be approximately 224 billions. The percentage difference between these figures is not large and is accounted for by the following:

1. More people are working than had been included in the estimate.
2. The shift toward manufacturing has been greater.

Analyzing these figures, we find that the sum of government expenditures plus foreign trade—which, in this case, is largely government-sponsored lend-lease or food relief—are very nearly the figures forecast.

Summation of the categories included in business also gives nearly the estimated figure.

All of the increase in gross national product, therefore, appears in consumer categories. Consumer durables are very nearly as predicted. Also, consumer services, including rent, are up only a little.

We now see how the realization differs from the prediction, and the question arises whether we have established a new pattern, or the return to the predicted pattern is just not yet complete. The realization of our economic potential is still seriously affected by material shortages.

Figure 3 lists 18 items in short supply with estimates of the probable dates when the shortages may terminate. One-half of the shortages are estimated to end in 1948, but the remainder, including such important items as fuel, grain, and steel, may be with us well into 1950.

In spite of these shortages, the economy is operating at high speed, is

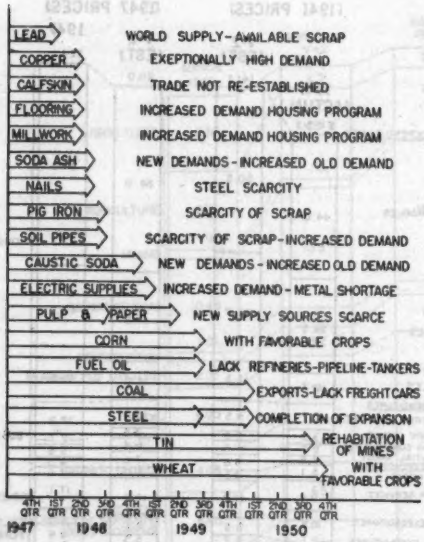

FIG. 3 SCARCITIES

producing record quantities of many items, and is reasonably in balance on the production side. Nevertheless, there is a great hue and cry that monopolists, taking advantage of the shortages, are reaping exorbitant profits. Figure 4 throws some light on that subject.

Who Reap the Fruits of Productivity?

In this chart gross national product is divided between government receipts, business net receipts, and individual net receipts. We see that over the years, while government's share has increased over 80 per cent, business' share has gone down 30 per cent and individual's share only 6 per cent. This does not sound like exorbitant profit increases, and the relative figures of corporate to individual receipts do not look excessive.

Moreover, Figure 5, further subdividing the receipts of individuals, shows that the share in wages and salaries did not go down at all but rather went up, as did the incomes of farmers and professional men. The losers were the recipients of interest, dividends, and rent. It is clear that business and the recipients of dividends, interest, and rent are the people with a justifiable complaint.

However, returning to the make-up of the "gross national product" and particularly to the expenditures of individuals, we find, as stated before, that consumer durables are about as predicted (see Figure 6). They constitute between 10 and 12 per cent of the "gross national product" and are within those limits now. Savings are still above normal rate; but, services, including rents, are off nearly one-third. A disproportionate amount of effort is going into nondurables. In Table 1, the distribution of the consumer dollar is further detailed. Only two radical departures from the norm appear, and these are rents, which are down, and food which is up.

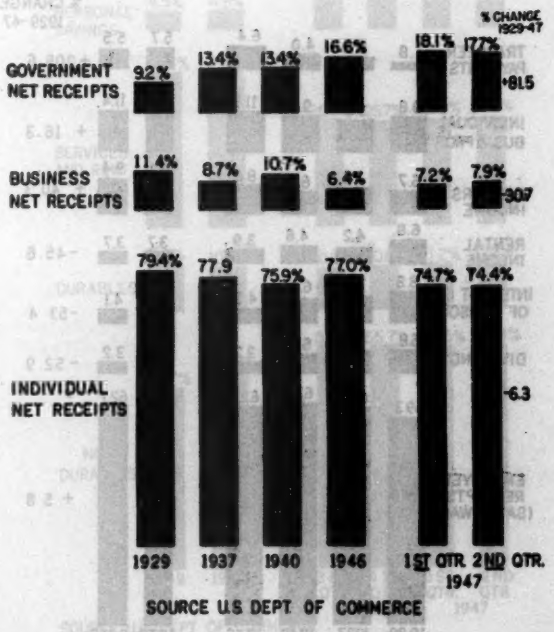

FIG. 4 PERCENTAGE DISTRIBUTION OF GROSS NATIONAL PRODUCT

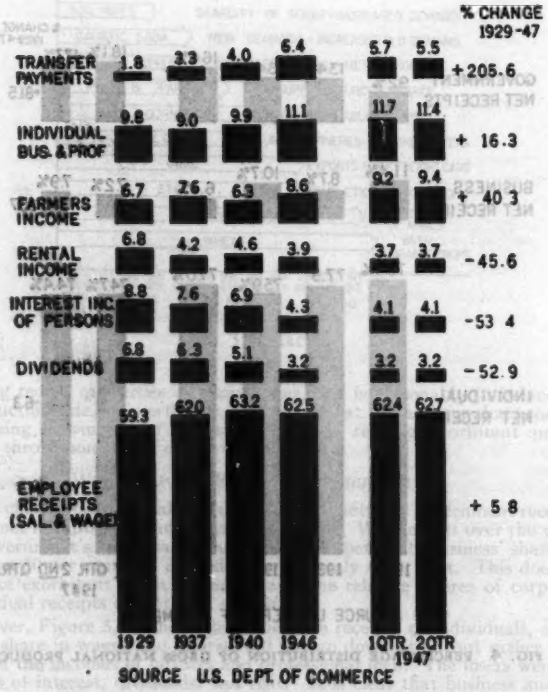

FIG. 5 PERCENTAGE DISTRIBUTION OF PERSONAL INCOME

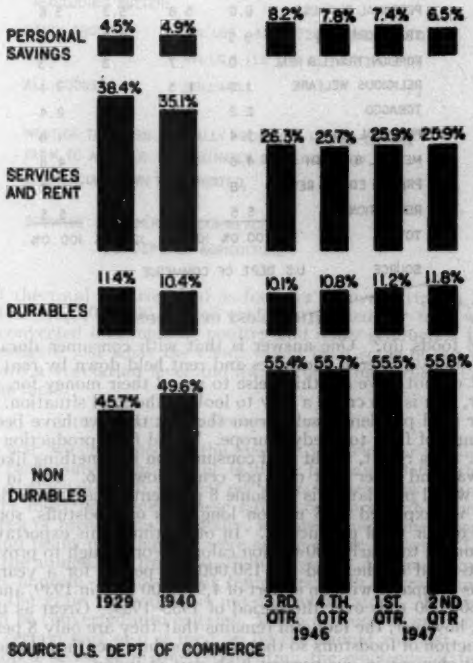

FIG. 6 PERCENTAGE USE OF DISPOSABLE INCOME

TABLE 1 PERCENTAGE DISTRIBUTION PERSONAL CONSUMPTION EXPENDITURES

	1929	1937	1940	1946
FOOD	25.0	29.7	28.8	36.0
HOUSING	14.5	12.5	12.8	8.8
HOUSEHOLD OPERATION	13.4	13.9	14.3	12.5
CLOTHES & ACCESS.	14.0	11.9	12.2	15.4
PERSONAL BUSINESS	6.6	5.8	5.3	3.6
TRANSPORTATION	9.5	9.6	9.7	7.6
FOREIGN TRAVEL & REM.	1.0	.7	.3	.3
RELIGIOUS WELFARE	1.5	1.3	1.4	1.1
TOBACCO	2.2	2.5	2.6	2.4
PERSONAL CARE	1.4	1.4	1.5	1.6
MEDICAL & DEATH	4.6	4.8	5.0	4.5
PRIVATE EDUC. & RES.	.8	.9	.9	.7
RECREATION	5.5	5.0	5.2	5.5
TOTAL	100.0%	100.0%	100.0%	100.0%

SOURCE U.S. DEPT. OF COMMERCE

High Cost of Foods.

Why are foods up? One answer is that with consumer durables held at present limits by material shortages and rent held down by rent control, the people just do not have anything else to spend their money for.

However, this is too crude a way to look at the food situation. Obviously, part of our food problem results from the fact that we have been exporting large amounts of food to needy Europe. World food production is definitely subnormal. As a result, world food consumption is something like 10 per cent below prewar and 2 per cent or 3 per cent below 1946. Yet, in spite of war losses, the world population is up some 8 per cent since prewar days.

In 1946 we exported 18.5 million long tons of foodstuffs, something like 8 per cent of our total production. In other times this exportation of foodstuffs amounted to nearly 500 trillion calories—or enough to provide approximately one-third of the food for 150,000,000 people for a year. This 18.5 million tons compares with an export of 4,900,000 tons in 1939, and an average of only 3,600,000 tons over the period of 1935–1939. Great as these exports have been, however, the fact still remains that they are only 8 per cent of our total production of foodstuffs so that, large as they are, they do not constitute a dominant element in our present high cost of living.

Although we are only proposing to export approximately 8 per cent of the total food production of the country, nevertheless the absolute amount is very large and, coming on top of very large domestic demands, it supplies a serious strain upon our economy. The total 1947–1948 grain supply is estimated at 6,253,000,000 bu. Of this total amount, approximately 900,000,000 bu. will be required for industry, seed, and carry-over. A total of 800,000,000 bu. are earmarked as food for domestic consumption, and some 4,083,000,000 bu. are earmarked as feed for animals. This leaves a total of 470,000,000 bu. of grain available for export. The estimated export requirements, however, are 570,000,000 bu., or a deficit of 100,000,000 bu. If we endeavor to get this extra 100,000,000 bu. by bidding in the market, obviously the effect on prices would be very inflationary, even though the total figure is only a little over 10 per cent of the total grain supply. On the other hand, the fact that almost

TABLE 2 UNITED STATES EXPORTS

COMMODITIES	UNITS	EXPORTS 1939	EXPORTS 1947 EST	% OF TOTAL PROD. EXPORTED 1939	% OF TOTAL PROD. EXPORTED 1947 EST
WHEAT	MIL.BU	77	392	10.4	27.3
CORN	MIL.BU	32	88	1.2	3.2
MEAT PRODUCTS	MIL.LBS.	193	328	1.1	1.4
EDIBLE FATS (EXCLUDING BUTTER)	MIL.LBS.	228	436	6.6	8.2
DAIRY PRODUCE	MIL.LBS.	41	728	4.1	49.0
COAL	MIL.L.G.T.	13	48	3.3	7.5
ALL GOODS	DOLLARS				7.1

MOST OF THE CORN NORMALLY PRODUCED IS FED ON THE FARM TO ANIMALS. APPROXIMATLY 25 PERCENT OF THE COMMERCIAL CORN IS EXPORTED.

SCOURCE: U.S. DEPT. OF COMMERCE
U.S. DEPT. OF AGRICULTURE

two-thirds of the total is earmarked as food for animals attracts immediate attention. The 4,083,000,000 bu. of grain earmarked for feed would presumably be converted into meat or poultry and dairy products. It is worth while to investigate just what the effect would be if some of our diets were directed toward a vegetarian level and away from a meat diet. Let us look at some of the facts.

Our Increasing Food Consumption.

Before the war, we consumed on the average 3250 cal. per person per day. Now, we are consuming at the rate of 3450 cal. per day. Before the war, we consumed an average of 2.6 lb. of meat per week, whereas now we are consuming 2.94 lb. of meat per week. These figures compare with approximately 4.2 lb. per week of breadstuffs. Actually, we, are eating some 8 per cent less of potatoes and beans now than we were eating before the war. To produce 1 lb. of beef, the animal eats from 6 to 10 lb. of grain. A pound of pork represents 4 to 6 lb. of grain. A pint of milk represents 1.7 lb. of grain; an egg, 9 oz.

Of course, the grain equivalent is not the whole story. However, these figures can all be converted into calories. When a pound of grain is eaten by an animal, a certain part of it is used up by the animal and lost. Therefore, we may set up a series of figures representing the efficiency of feeding the human being by way of grain coming to the human being through different channels. Of our common foods, the most inefficient in this respect is butter. For each calorie we consume as butter, some 39 times as much grain has been expended in the form of grain fed to the cow. This is even after giving due credit to the calorific value of the milk which, when not used as butter, is refed to animals. A chicken consumes 18 cal. for each calorie we get from his meat. An egg represents 13 cal. for every one which we absorb. For beef, the figure is 9; for milk, 6; and for pork, 5. Thus, the pig is the most efficient converter of grain to meat and, even in the case of the pig, 80 per cent of the food value of the grain is lost in the process.

Now, since two-thirds of our grain supply is fed to animals—and since, in even the most efficient case, 80 per cent of the food value of the grain is lost in the conversion—the most practical way to stretch our world food supply is to reduce the amount of these animal products which we consume and

thereby release the corresponding grain for direct human consumption. Most of the pressure so far has been in the direction of urging people actually to consume less. We in the United States are very well fed compared with practically all other areas of the world. However, any reduction in the calorific input is accompanied by a certain amount of hazard. Well-fed people are much more immune to various kinds of sickness than are those who are on a semistarvation diet. In any event, it does not seem practical to urge the people of the United States to starve themselves in order to help out the starving peoples of Europe when no such step is actually necessary (at least not this year).

Dietary Suggestions.

To demonstrate this point, we asked a dietitian to prepare two alternative menus. The first represents an amount of meat, eggs, butter, and cream reasonably close to the national average per capita consumption of these products. The second shows a lower meat diet in which careful attention has been given to the proper amounts of all vitamins and minerals. This low meat diet is still above the minimum protein standard required. The total calories provided are actually higher than in the high meat diet. These menus, together with their calorific and other content, are shown in Table 3.

If the whole population were to shift over to this low meat diet, it is possible to show that the amount of grain realized would be about 2,600,000,000 bu., 6 times the total grain exported last year, or 4 times the desired total of grain exports for this year. Under these conditions, 2,600,000,000 bu. of grain would be saved out of the total of 4,000,000,000 bu. which it is estimated will be fed to animals this year.

Such a drastic change is clearly impractical. In addition to changing the diet of a whole nation—if only 1,400,000,000 bu. of grain were required—obviously, farmers would not raise 4,000,000,000. Not only would such a change upset grain production, but it would also drastically reduce the amount of fats, hides, and other products available which are required by industry.

The problem, therefore, is to make a shift toward a lower meat diet sufficient to make available enough grain to supply our foreign-aid requirements. Here are some examples which show the amount of shift which would be required.

Before the war, most cattle and hogs were slaughtered when they had attained an economical marketing grade. This grade, often known as "good slaughter grade," was such that the maximum proportion of the fats was consumed by human beings. Today, cattle and hogs are being slaughtered when they have attained a higher grade, often called "choice grade." Much of the additional fat which these animals carry is not consumed by human beings, and in the production of these fats the use of grain is ⅓ less efficient than when the animals are at "good slaughter grades." It is estimated that, by marketing cattle and hogs at prewar weights, we would save about 200,000,000 bu. of grain.

Before the war the per capita consumption of meat was 133 lb. per annum—today it is 153 lb. A saving of 20 lb. per capita per annum would release about 350,000,000 bu. of grain.

Thus these two measures alone would provide enough grain to meet any foreign food deficiency we might wish to overcome. In addition, it would mean that Americans would be called upon to reduce their consumption of meats by 15 per cent, still leaving them far ahead of any of the less fortunate people in other countries.

If we were to reduce the amount of milk consumption of the country only 1 pt. per person per week, we would release approximately 200,000,000 bu. of grain per year, or twice the amount of the indicated deficit. It is clear, therefore, that the way out of the world's food dilemma is not in the direction of belt-tightening but is rather in the direction of going from a high meat to a somewhat lower meat diet.

TABLE 3—TYPICAL MENUS AND THEIR CALORIFIC VALUES.

Average Meat Menu	*Low Meat Menu*
Breakfast:	Breakfast:
Grapefruit half	Stewed prunes
2 Eggs with bacon	Oatmeal with milk and sugar
2 Slices toast with butter	2 Slices toast with margarine and jelly
Coffee, cream and sugar	Coffee, top milk and sugar
Lunch:	Lunch:
Cream of pea soup	Cream of tomato soup
Swiss cheese sandwich	Peanut butter and jelly sandwich
Boiled ham sandwich	Egg-salad sandwich
Fresh orange	Fresh orange
Glass of milk	Glass of milk
Evening meal:	Evening meal:
Roast beef au jus	Boston baked beans
Baked sweet potato	Broiled bacon
Fresh frozen asparagus tips	Fresh spinach
Hot roll with butter	Tomato-and-lettuce salad with French dressing
Ice cream with chocolate sauce	2 Slices bread with margarine
Glass of milk	Gingerbread with apple sauce
	Glass of milk

	Standards	*Low Meat Menu*	*Average Meat Menu*
Protein	47–105 g	90 g	124.0 g
Carbohydrate	280–420 g	402.9 g	312.0 g
Fat	70–140 g	166.6 g	182.6 g
Calories	2800–3500	3388	3349
Calcium	.8–1 g	1.323 g	1.308 g
Phosphorus	1.32 g	1.902 g	1.964 g
Iron	12 mg	34.6 mg	12.9 mg
Vitamin A	3000–5000 I U	26441 I U	10461 I U
Thiamine	2.3 mg	3.96 mg	2.559 mg
Vitamin C	75 mg	139.7 mg	161.9 mg

NOTE: The figures used are based on the daily requirements of a 70-kg (154-lb.) man. The variance allowance in requirements is dependent upon the activity of the man. A man of this weight uses from 4–5 cal. per kg of body weight depending upon his activity. His protein variance is from ⅗–1½ g of protein per kg of body weight. The carbohydrate variance is from 4–6 g of carbohydrate per kg of body weight.

Actually, since the data for this paper were first brought together, some change for the better appears to have taken place. Apparently, farmers have not been feeding as much grain to animals as was originally estimated, and it appears as though the 570,000,000 bu. desired for the 1947–1948 aid program are already on hand. However, the foregoing analysis indicates the direction and extent of any sacrifices which the United States might have to consider in the event that such a program of aid was needed another year and should our own harvest prove to be less bountiful than it has been in the last six years.

But relieving a world food shortage is not the only reason for proceeding in the direction of a lower meat diet. In Schenectady the cost per day of the food in the high meat diet is approximately $1.70; that of the low meat diet is approximately $0.90, or only slightly more than one-half as much; and this saving assumes no change in prices.

How to Bring Down Living Costs.

We say we wish to bring down the cost of living. Here is the way to do it. Even if the prices of the articles should remain the same, the use of substitutes for grain-eating animals, such as sheep, fish, or protein vegetables, including many varieties of beans and peas, would bring down living costs. If enough of us reduce our consumption of grain-eating animal products, surplus grain will come on the market and the price will go down. With it will go the price of such meat as we do eat.

But, to return to our major thesis, the totalitarian way to abundance has been tried and failed. Stalin, where he succeeds, is using modified capitalistic methods. Socialism in England is not making a very enviable record. Our own efforts at government intervention have not been entirely successful.

Employment and Production Depend Upon Competitive Enterprise.

The only really successful method of achieving high-level employment and production has been the method of a competitive-enterprise system. In the past, that method has been subject to periodic recessions. It is believed we now know how to avoid serious recessions. As a result of postwar planning by businessmen, largely inspired by the Committee for Economic Development, we have avoided the recession that many people inside Government and out were sure would engulf us before now.

The principle of the thing is simple. Estimate what it is thought the country can produce in total and what things the people with the purchasing power will probably choose to buy of free choice. Make and sell those things. If it were not for the labor and detail necessary to elaborate these simple principles, this would sound too easy. Actually, such estimates have never been possible before because of lack of information. The information is available now. It has worked well enough to help us reconvert and get into all-time high-production territory. Now we must steer a course in which we remain faithful to the methods that got us where we are but not get overconfident. We can proceed with confidence, but we must watch for signs of danger.

One obvious sign of danger is an excessive accumulation of inventory in relation to sales volume. Some signs of this character have appeared already. Table-model radios are a good example. Certain makes of refrigerators and washing machines are also in ample supply. While manufacturers of these products, as a whole, are still limited by the availability of materials, mostly steel, all manufacturers are on notice that the rate of production is approaching adequate proportions.

However, the successful application of the relatively simple principles involved in continued high production and employment involves organizing to anticipate a condition of adequate production in advance of the time when excess stocks appear in the store windows. By that time, the only effective course to pursue is to cut production.

Much more success is to be anticipated when it is possible for the majority of concerns to anticipate the likelihood of this condition arising some months in advance of its actual occurrence. Much work needs to be done before such principles are generally applied throughout industry, but large corporations are already building up staffs of economists and market analysts to watch markets and give warning at as early a point as possible of potential overproduction. Much good can also be done by expanding the statistical activities of trade associations so that more data are made available to members at an early date. Finally, government agencies such as the Department of Commerce and the Department of Labor should be encouraged to improve their methods of collecting and presenting data. In addition to the already increasing mass of improved statistics which these agencies provide, private agencies are covering specialized fields by sampling methods.

Statistical Business Studies Needed.

The project, as a whole, involves a new large-scale problem in communications. It requires the rapid collection, digestion, and dissemination of business information in forms which can be comprehended readily and applied by businessmen to the everyday problems of selling, production planning, and inventory control. It is likely to require a co-operative effort between businessmen, analysts, and engineers to devise the systems by means of which the necessary information can be collected and disseminated with the least disturbance to the ordinary processes of business.

In any year, the total purchasing power created and available to individuals, business, and government is exactly equal to the value of the goods and services produced over-all. To maintain full employment, it is necessary that the specific goods and services produced by those which the receivers of the purchasing power wish to buy.

This increasing mass of data is making it more and more possible for the managers responsible for production to direct it through the channels in the proportion that it will be bought by those receiving the purchasing power. In this way the individual purchaser is king.

Managers and labor thrive in proportion to the skill management uses in predicting the wishes of the customer and in selling him on the idea that his standard of living should go on rising. That, in effect, is the pattern of our competitive-enterprise system. That is the system which has brought us where we are with the highest standard of living of any country in the world. It is the method most likely to keep us there.

RADIOGRAPHY AS A CONTROL FOR WELDING JOINTS IN PIPE LINES.

This article by Russell G. Rhoades is reprinted from "The Welding Journal" for June 1947.

At times we may ask ourselves—why do we work for industrial progress? Why do we work for ever-increasing production efficiency? The answer to this query may lie in the fact that constant recurrence of change acts as a stimulus and a challenge. However, it is our fulfillment of this industrial efficiency in our work each day which distinguishes us as American individuals in our daily standard of living over any other group of people on this earth.

It is human tradition to keep close to the beaten path and live today by the habits of yesterday. Someone has said, "At times we must take a pole to drive a cow to new pastures." However, we are always proud of ourselves when we find that we, by some good common sense and reason, have turned from our old ways and have given ourselves to the new. New uses for old

techniques are constantly being discovered by engineers and technicians in every line of endeavor. This paper deals with the new use of a previously known technique with some adaptations.

Radiography is not new in its use to inspect and control welding, but it is new to use radiography on a pipe line where groups of construction men work with the force and speed necessary to weld, treat and complete the laying of 4000 to 10,000 ft. of line per day.

Theories and processes developed in the laboratory have sometimes proved impractical when put to actual field use. However, in this case we actually used the methods, to be described later, on the construction of 176 miles of 18-in. high-pressure oil line for a major oil company—The Standard Oil Company of California.

The vistas opened by the development of this new technique are limitless; however, this report shall be confined to its use in pipe line work only.

First, both the customer and the builder receive benefits when each takes the available advantages which radiography offers in constructing an all-welded pipe line. This means there must be close coordination between the customer and the builder in their efforts to build a pipe line with predetermined joint specifications.

The inspection department of the customer and the welding supervision of the builder will work in common understanding when together they have adequately tested enough joints to establish welding standards. When the test results of any welding procedure reveal sufficient and substantial evidence—only then a standard of welding quality in detail can be determined.

Since using radiography in constructing a pipe line is relatively new and considering the kind of men who lay pipe lines, we see the importance of providing a common working knowledge of radiography—what a radiograph reveals coordinated with results of physical weld tests.

The prime objective which we must accomplish in welding any pipe line is to weld a joint of 100 per cent efficiency in all the physicals to which a pipe line may be subjected in the course of its life. A 100 per cent joint in a pipe line is composed of approximately 90 per cent tensile with the 10 per cent remainder found in ductility. A pipe line joint seldom requires attention to such physicals as torque, shear and fatigue to any appreciable degree. We still would have some difficulty welding all the physicals in the weld deposit perfectly and identical to the parent pipe. If we should weld all the physicals equal to the parent pipe, in size, thickness and shape, virtually we would have produced seamless pipe.

We must control the elements which give us a homogeneous welded joint with the least weld deposit. After we have established the necessary physicals of our procedure, the shortcomings we must minimize in acquiring both construction efficiency and joint efficiency are as follows:

1. Misalignment of pipe ends.
2. Lack of penetration.
3. Burned through areas.
4. Slag inclusions (gas pockets).
5. Undercuts (outside, inside).
6. Size and shape of weld (outside and inside—especially inside).

To minimize these deficiencies in welding a pipe line we continue to use several established ways of inspection. Most of these ways are clumsy and costly. They are usually operated with awkwardness and they are greatly lacking in thoroughness and accuracy.

Radiography is steadily being found as a new and more simple means to quickly recognize and evaluate the above-described shortcomings. We can see build-up and undercut on the outside with the naked eye, but the joint cannot be reliably evaluated until it is considered with the condition on the inside of the pipe. Radiography has no equal in providing the facts of these welding conditions in a pipe line.

To provide sufficiently powerful mobile X-ray equipment behind the fast working pipe line welders and yet preceding the treating and line-laying crews, would involve prohibitive transportation costs.

Radium in capsule form is an excellent light source because it is self contained in a capsule. These capsules are small, being approximately ¾ in. in diameter and 1 in. long, and often shaped like a pear or a schoolboy's top.

In using either X-ray or Gamma ray, when both the light source and film are on opposite sides of the pipe, other disadvantages are always present, such as producing an elipse radiograph. The light must penetrate the obstructing opposite side wall of the pipe before it can reach the small portion of the weld on the other side. This is especially a disadvantage with big-inch pipe, and double wall thickness, of course, requires double exposure time. The distances from the light source to the various points on the film are unequal which hinders uniform film exposure. By the angle of the lines by which the light travels from the source through the weld and reaches the film in its circular position outside the weld, the image on the film is produced in non-uniform distortion.

Our objective and our accomplishment has been to place this light source on the inside of the pipe, and the film around the outer periphery of the pipe wall. This is achieved by cutting a small hole in the pipe near the joint to be tested, and inserting the radium capsule inside by means of a specially constructed self-aligning jig. This could not be done with cumbersome X-ray equipment. Placing the radium or light source on the inside of the pipe provides the following advantages:

1. The complete circumferential weld can be examined with a single exposure. All the film is equidistant from the light source.
2. Distance from light source to film is reduced, thus exposure time is reduced by the square of the distance.
3. The single wall thickness, compared with two through which the light must pass, also reduces the exposure time by half.
4. All the radiograph of the weld on the film is exposed to scale and without distortion or any elipse. This is a factor which provides accurate detail.
5. Considering the direction of the lines which shape the full bevel or V from the outside, further advantages are gained in radiographic detail when the film is placed on the outside and light source is on the inside, in contrast to vice versa.

With 300 milligrams of radium, using fine grain film and proper screens on 18-in. pipe with quarter inch walls, an exposure can be made in 12 to 14 min. The film may be developed and read in 15 to 30 min. after it has reached the darkroom.

The hole cut in the pipe is welded by using a special inverted spigot plug. Engineering actually shows that a ¾-in. hole in a large diameter pipe does not weaken the over-all strength of the pipe. In fact we did not experience a single leak by using this type of plug with the full fillet weld, and this procedure was highly satisfactory to both customer and the engineers.

These radiograph methods with the hole as we have described are in the specifications of a portion of the Texas Los Angeles 30-in. gas line.

On the job to which I have previously given reference were employed the Industrial X-Ray Engineers of San Francisco. By close coordination between the X-Ray Engineers, the customer and the contractor a set of standards and specifications were established.

We accomplished this end by welding 20 pipe joints each of which was known to include one of the following conditions: misalignment, lack of penetration, burn through areas, gas and slag pockets, undercuts, different weld sizes and shapes, and perfect welds. These joints were radiographed and indexed as to location and irregularity. Coupons from the above 20 joints were given physical tests after being cross indexed with the radiograph film. The results were then correlated with our findings on the film to be used as a basis for a set of standards. This whole procedure was meticulously carried out, and any uncertainties were retested and rechecked.

Because of the accurateness and regularity with which these radiographs provided in a daily record of the type of joint being welded in the pipe line and when compared to the established standards, the work had scarcely started before radiography had proved itself indispensable to both the customer and the contractor.

It is generally known that through inadequate or negligent handling, radium is not without its dangers. This is one reason trained engineers should be employed to use it on a pipe line.

Due to the extreme mobility required on a pipe line job, a house trailer as a combination laboratory, darkroom and office for compilation of records was used. A triplicate record was maintained of all welds radiographed.

A system of numbering the welds and locating the welds in the pipe line was radiographed on the film. Also a similar clock-wise numbering system registered on the film the exact location of any irregularity or defect in the circumferential weld.

When welders understand how radiography works they like it and they appreciate it in that it assists them to make a more perfect weld.

The number of radiographs required to control the welding on a pipe line depends upon the size of the line—the number of daily welders, etc. However 5 to 10 per cent of the welds made are found to be adequate to provide dependable information.

The common disadvantages encountered in examining pipe line welds by the destructive methods are as follows:

1. The costs of replacing the section cut out for testing.
2. The machinery to prepare coupons and the equipment for testing is seldom readily accessible to a pipe line.
3. The natural forces tend to reduce testing to a point where a working knowledge of welding and joint value is practically out of the hands of the daily workmen and inspectors.
4. With the destructive tests, as we have commonly used in the past, we can only test the weld which we take out of the pipe line.
5. The tested weld never stays in the line.

With radium inspection, the welding force is daily informed of the standards by which they welded the joints in the line. And if there are any defects they know exactly what to do to correct the irregularities. The customer holds a permanent record of the tested welds in his pipe line. The builder knows the standard of joints he welded in the pipe line. The customer knows what kind of a pipe line was welded and what he receives.

The basis for success in using radiography on a pipe line or any similar welding job, is that both the builder and the customer must realize there is a benefit and a saving for each, then further realize that a way must be provided whereby each will obtain his just portion.

Such critical information as provided by radiography must not be used by one to the disadvantage of the other, but must benefit all concerned. Gains will be more plentiful to all if radiography is accepted in the early planning of the welding project.

Briefly I have endeavored to cover the high lights of a new technique in welding analysis. I would summarize as follows: Mere mechanical processes and innovations are not enough to fully accomplish the full purpose. Radiography, however, provides more welding and joint information than ordinarily obtained from common methods of investigation. Efficiency is not attained as a result of uncertainties, or from common statements and beliefs. Instead, efficiency in pipe line welding, as in any other work, is found at its highest level only when all information and details available are applied to the fulfillment of that work.

The more we control the weld deposit in acquiring the needed physicals, the nearer the deposit will equal the pipe in thickness and shape. With radiography we raise two efficiencies—one in the joint, the other in our way to make it.

BOOK REVIEWS.

INTRODUCTION TO THE DIFFERENTIAL EQUATIONS OF PHYSICS, By L. Hopf, Professor of the Aachen Institute of Technology, (Translated by Walter Nef, Professor of the University of Fribourg, Switzerland). Published by Dover Publications, Inc., New York, 1948. 152 Pages; $1.95.

This small volume emphasizes the intuitive rather than the formal aspects of mathematical developments. It stresses primarily the view of the physicist or engineer. An immediate understanding of the subject goes far beyond what might be expected from the size of the volume.

There are seven chapters:

I. The differential equation as an expression of a law of nature.
II. The ordinary differential equations of the mechanics of particles.
III. The simplest partial differential expressions.
IV. The simplest partial differential equations of physics.
V. Solutions of Eigen functions.
VI. Solution by change of variables.
VII. Solution by the use of singularities.

ELEVEN AND FIFTEEN PLACE TABLES OF BESSEL FUNCTIONS OF THE FIRST KIND, TO ALL SIGNIFICANT ORDERS, By Enzo Cambi, Eng. D. Published by Dover Publications, Inc., New York, 160 Pages; $3.95.

This volume meets a persistent demand for a handy collection of tables of Bessel functions of high integral orders. A range, fully adequate for most applications in physics and engineering, is provided by the main table which gives $Jn(X)$ for $x = 0(0.01)\ 10.5$ and $n = 0(1)\ 29$ to eleven places. A supplemental table gives $Jn(X)$ for $x = 0(0.001)0.5$ and $n = 0(1)11$ to fifteen places.

Table of Contents:

Preface.
Graphs of the upper limits of error.
Explanation of tables.
Taylor series of Jn of even order around x = 2, 3, 4, 5, 6.
Addition series of Jn of even order, x = 7, 8, 9, 10.
Bibliography.

A HISTORY OF THE UNITED STATES NAVY, By Commodore Dudley W. Knox, U. S. N., with a Foreword by Fleet Admiral Chester W. Nimitz, U. S. N., and an Introduction by Vice Admiral William L. Rodgers, U. S. N. Published by G. P. Putnam's Sons, New York. 632 Pages; $7.50.

For those familiar with the previous edition of Commodore Knox's History of the United States Navy, it need only be said that this edition has been brought up to date by the addition of 198 pages covering World War II.

The 632 pages of narrative with its 72 page index provides an encyclopedic summary of cause and effect as well as events which constitute the story of our Navy from the Continental period until today.

The reader is torn between two thoughts: regret that the author did not expand many portions of the book into greater detail and amazement that he accomplished in such narrow limits a history not lacking in any essential.

THE SHIPBUILDING BUSINESS IN THE UNITED STATES OF AMERICA. Volumes I and II. Edited by Professor F. G. Fassett, Jr., Director, Office of Publications, Carnegie Institution of Washington. Size 8 by 10¾ inches. Pages, Volume I, 313; Volume II, 250. Numerous illustrations, tables and charts. The Society of Naval Architects and Marine Engineers, 29 West 39th Street, New York 18, N. Y. Price $12.50 per set of two volumes, postpaid in the United States. Foreign postage—$1.25 per set.

Third in its series of basic publications having to do with the design, construction and operation of ships, the two volumes

entitled *The Shipbuilding Business in the United States of America*, under the editorship of Professor F. G. Fassett, Jr., of the Carnegie Institution of Washington, have just been issued by The Society of Naval Architects and Marine Engineers. The preceding titles in this series, *Principles of Naval Architecture*, issued in 1939, and *Marine Engineering*, which appeared in 1942 and 1944, respectively, were concerned with theory and practice involved in the design and construction of vessels. The present publication is unique in that it is the first serious effort on the part of a major industry in the United States to set out in sequence and detail the procedures and practices which have resulted in the successful functioning of the industry as a part of the American economic system of free enterprise. The thirty authors whose contributions were co-ordinated and unified by the editor under the general direction of a control committee of Society members are all experts drawn from various portions of the industry, each selected for his recognized knowledge of and ability to discuss the certain phase allotted him of this widely ramified business of building ships. As a consequence, the volumes assemble authoritative discussions not only of the broad historical and economic aspects of a basic industry, but also detailed explanations and analyses of successful operating procedures, drawn from the experience of major shipyards from all the important shipbuilding centers of the country.

The volumes are copiously illustrated with diagrams, figures, graphs, tables, and photographs. These include a comprehensive series of plot plans showing the layout of many representative shipyards, as well as many specimen illustrations of operating charts and tables used in the conduct of the business. Thus the illustrative materials valuably supplement the discussion in showing both the physical background of the problems of shipyard management and significant selections from among the methods used in solving them. The eighteen chapters making up the two volumes are carefully cross-referenced and interrelated, so that the complete work constitutes an integral unit.

The first three chapters of Volume I, treating the economics, general history, and specific factual statistical record of the shipbuilding industry in this country, at once orient the reader to insure his ability to appraise the remaining discussion. Chapter

I analyzes the special characteristics of shipbuilding as a basic element in the economic system of the United States, pointing out similarities and differences which enable the reader to appreciate this industry's relation to others and to distinguish the ways in which it is a unique business. Chapter II summarizes the history of the industry from 1776 to the present and provides a comprehensive survey which illustrates clearly from the actual record the economic analysis which has preceded it. Chapter III assembles most comprehensive statistical records of actual production and other business facts, including statistical summaries extending from 1789 to the present with explanations and discussions of the extensive tabulated material.

Chapter IV describes the physical layout of representative shipyards and the reciprocal influence of the type of construction on the shipyard design. This chapter sets out clearly the fundamental distinctions between "standard" yards building vessels of many different designs and "multiple-production" yards building large numbers of duplicate vessels—a difference which recurs in many other portions of the book. The system of organization of standard yards and multiple-production yards is considered in Chapter V, the functional organization of the staff varying as required by the types of ship construction and the plan of organization being in turn correlated with the various functional responsibilities. Labor in shipbuilding is the subject of Chapter VI. The rise of the union movement in shipyards and the problems engendered by the peak employment of World War I and World War II are chronicled and the various methods, industrial and governmental, employed to meet them are analyzed. The discussion of "Shipyard Wage Systems" in Chapter VII presents pro and con arguments for and against the day's rate and the wage incentive plans and incorporates the results of extensive inquiry into existing practices in the different shipyards.

Having defined and discussed in Volume I these broad elements of the industry, *The Shipbuilding Business in the United States of America* proceeds in Volume II to describe in logical sequence the exacting detailed managerial and operational functions which must be performed for the efficient and economical construction of ships. Chapter I of Volume II starts the story

by discussing procedure involved in the seeking of proposals to build and the completion of shipbuilding contracts. Sections on United States Navy procedure, United States Maritime Commission procedure, and commercial procedure, each by an expert in the field, not only explain specific contract requirements but recount the way in which they have been developed and the reasons underlying them. Chapter II in similarly logical order considers the vital functions of planning, designing and scheduling. The critical importance of correct erection sequencing is clearly explained in this chapter, followed by detailed analysis of cost estimating in Chapter III where methods used to determine probable cost are fully described. Chapter IV is concerned with production and material control, its two sections covering the point of view of the standard yard in one and that of the multi-production yard in the other. These sections are carefully correlated, so that characteristics common to both and those distinctive of each are pointed out.

The succeeding three Chapters V, VI and VII discuss procurement and storekeeping, shipyard costkeeping and accounting and corporate accounting and management controls, their subject matter being closely integrated with the commentaries on labor costs and policies made elsewhere in the two volumes. The function of expediting is here discussed, and the close relationship between costkeeping and accounting and the general problem of material and production control is analyzed. Charts, graphs, and tabular illustrative material contribute substantially to the effectiveness of these chapters.

Chapter VIII treats the subject of marine insurance from the point of view of the shipbuilder. Its two sections deal respectively with insurance on builder's risks and with ship repairer's liability insurance, subjects of extreme importance to the industry infrequently considered with the thoroughness characteristic of this chapter. The cognate subjects of "Classification Society Rules and Interconnection of Government and the Shipping Industry" and "Shipbuilding Inspection" are covered in Chapters IX and X. The evolution of the classification society and the reliance placed upon it by government authorities are explained and the international character of shipbuilding operations is summarized and the various forms of inspection performed by

the numerous agencies concerned with a vessel's construction are described. The Volume concludes with Chapter XI, which compares shipbuilding and ship repairing procedure, Section 1 dealing with private repair yards and Section 2 with Navy Yards. The essential difference between the problem of construction and that of repair is stressed in each section and the resulting difference in method of attack and technique imposed are clearly described.

This latest addition to the Society's list of basic treatises in its field is a logical and satisfactory successor to the earlier volumes. It will be of interest and value to students and management in many industries other than shipbuilding.

ERRATA.

On page 189, line 26, of the paper "The Navy in Research," by Captain William H. Leahy, U. S. N., in the May, 1948, issue of the JOURNAL, "−459.72° F." should be substituted for "−494.72° F.".

ERRATA.

On page 189, line 26, of the paper "The Navy in Research," by Captain William H. Leahy, U. S. N., in the May, 1918, issue of the JOURNAL, "−459.72° F." should be substituted for "−454.72° F.".

ASSOCIATION NOTES.

CANONS OF ETHICS FOR ENGINEERS.

"Canons of Ethics for Engineers," which was published in the notes of the May, 1948, number of the JOURNAL, is eminently complimented by the following quotation from an editorial which appeared in *Mechanical Engineering* (December, 1947).

"Few if any of the contributions by engineers which give expression to the spirit of public service which animates the engineering profession at its best have been read with approval by more persons than (1) Wickenden's 'The Second Mile,' an address in which he shows that professionalism in engineering lays on its practitioners the obligation of rendering more service than duty demands. To quote his own comments in this remarkable address:

'One of the most suggestive verses in the Bible is this: 'If a man compel thee to go one mile, go with him twain.' This text obviously is a counsel of perfection, foolish if taken literally, but emphasizing a profound truth by its apparent denial of common sense. What it means is that many of life's experiences begin with a mile of compulsion but that its true freedom and durable rewards are to be won only in the mile of striving which lies beyond. In the first mile men must work to live; in the second they work to maintain their sense of dignity and worth. In the first mile they seek tangible rewards; in the second they strive for enduring satisfaction. In the first mile they seek pleasure; in the second they discover happiness. The first mile is the mile of discipline; the second is the mile of freedom.'

'Real freedom,' wrote Wickenden elsewhere, 'is not a birthright; it is something you have to win the hard way. No man's education is even well begun until he can do something extremely well. It does not matter so much what—type a letter, bake a pie, weld a seam, paint a house, do a portrait, beat a drum, compose a symphony, operate a radar, or design

(1) W. E. Wickenden, President of Case Institute of Technology until his death last summer.

a Boulder Dam. What counts is the self-mastery that enables a man to look the world in the eye with steady self-assurance.

'The adequacy of any educational plan or system must be weighted in the scale of citizenship, no less than that of personal self-mastery, of career fitness, of cultural orientation, and of social and group morality.'"

ADDRESSES.

Addresses are still a source of much concern. It is again urged that each member keep the Society informed as to changes.

ANNUAL BANQUET.

The Annual Banquet of the Society was held at the Statler Hotel, Washington, D. C., on May 7, 1948. Dinner was served to 1131 members and guests, who constituted the largest number it has been the privilege of the Society to greet on any of these occasions to date. While the event taxed to the utmost the available facilities both for the dinner and for accommodations for entertainment parties and for sleeping, it was again an outstanding success.

Rear Admiral Roger W. Paine, U.S.N., President of the Society presided and welcomed the guests. Rear Admiral Clinton E. Braine, U.S.N., served most acceptably as Toastmaster. Splendid addresses were given by Vice Admiral William M. Fechteler, U.S.N. entitled "Amphibious Operations in New Guinea and the Philippines" and by Dr. Detlev W. Bronk, Chairman, National Research Council, whose subject was "Science and National Survival".

A delightful musical program was rendered by an orchestra from the U. S. Marine Corps Band, under the leadership of Major William F. Santelmann, U.S.M.C.

During the evening, Rear Admiral Paine presented to Commander George C. Wells, U.S.N., and to Mr. Albert F. Bird the awards made to them in the Prize Essay Contest.

Notwithstanding current extremely high prices, the Banquet Committee, with the approval of the Council, determined to maintain the standard of the dinner without increasing the cost to members. As was anticipated, this procedure resulted in the banquet not being self-supporting. The practicability of continu-

ing to incur a deficit in this connection will be the subject of consideration in the future. Care will, of course, be exercised to ensure that in this activity, as in all other transactions of the Society, profit shall not be contemplated and that a maximum of entertainment shall be offered.

Scholarship.

Two applications only were received for the Scholarship offered by the Society for 1948. This scholarship has been awarded to Thomas Gillette, Stepson of Rear Admiral Claude S. Gillette, U.S.N., Retired, Member.

Change in Council.

Due to his retirement and removal from Washington, Rear Admiral J. W. Fowler, U.S.N., has resigned as a member of the Council and Rear Admiral Theodore C. Lonnquest, U.S.N., has been appointed in his place to serve until December 31, 1948.

Society Lapel Button.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a very fine dignified insignia. It is one-half inch in diameter.

The oak leaves and lettering are red on a gold back-ground. It is available to all members at fifty cents (50c) each.

Membership.

Attention of all non-member officers is especially invited at this time to the advantage and benefit to be attained by membership in this Society. The Society was organized in 1888—for the sole purpose of the advancement of Engineering. Its

JOURNAL, published quarterly, in February, May, August and November, now in its 60th year of continuous publication, has attained a foremost rank in its field and is a recognized authority throughout the engineering world. It is used generally by Naval officers and civilians alike as a reliable reference book. The Bureau of Ships considers the JOURNAL of such value that it makes it available for the libraries of all Naval vessels and shore activities of any size. The presence of the JOURNAL in your personal library cannot fail to be of great benefit to you, often making available information of great value which can be obtained from no other source.

It is particularly suggested that officers of the Naval Reserve, who are returning to civil life, should arrange for membership prior to detachment from active duty.

Annual dues are $5.00, which are probably lower than those of any other technical Society. There is no initiation fee, nor any extra charge for the JOURNAL. This extremely low cost is possible only because the Society is bona fide operated solely for your benefit without any pecuniary profit to any one.

An application blank generally will be found at the back of each copy of the JOURNAL. However, if none is found, application by letter will be accepted.

It is suggested that Members will confer a real favor on non-members by bringing the Society to their attention.

MEMBERSHIP ADDITIONS.

The following have joined the Society since the publication of the February, 1948, JOURNAL:

NAVAL.

Bacheler, Theodore F., Lieut., U.S.N.,
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